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PRELIMINARY STUDIES OF A WHEEL PUMP FOR THE PROPULSION OF FLOATING VEHICLES

I. Robert Ehrlich, et al

Stevens Institute of Technology
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ABSTRACT

A novel propulsion device for an amphibous wheeled vehicle is described. This device, which is an integral part of the vehicle wheels, pumps water between the tire rim and the brake drum inboard into a stationary collector which turns the water rearward, thereby generating forward thrust.

Results of preliminary tests conducted on a stationary pumping system and when mounted on a M151 $\frac{1}{4}$-ton truck are presented.

Tests indicated that the device increases the maximum bollard pull approximately 100% and the maximum speed approximately 40% over propulsion with tires alone. It also materially improves the controllability of the vehicle.

KEYWORDS

Amphibians
Swimmers
Floaters
Propulsion

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NOMENCLATURE

A	Area
A_p	Effective Cross-Sectional Areas of the Pump
A_S	Vehicle Submerged Frontal Area
C_D	Drag Coefficient
C_{T_p}	Pump Thrust Coefficient
D	Impeller Diameter
E	Energy
H	Pressure Head Across Pump
HP_o	Output Horsepower
Q	Flow Volume
T	Thrust
U	Vehicle Velocity
V	Velocity
g	Gravitational Constant
n	Impeller Rotational Speed (rps)
p	Pressure
u	The Increased Flow Velocity Imparted by the Pump
v	Mean Flow Velocity Across Pump
δ	Specific Diameter
η	Efficiency
γ	Specific Weight
λ	$U/\pi nD$ = Advance Coefficient
φ	Capacity Coefficient
ψ	Pressure Coefficient
ρ	Density
σ	Specific Speed

INTRODUCTION

It has long been known to the designers of amphibious and floating vehicles that the addition of a screw propeller or waterjet is needed to achieve reasonable water speeds, as on the highly successful World War II DUKW, the amphibious Volkswagon, the LARC V and XV, and, more recently, the LVTP-X12. Unfortunately, waterjets and propellers add additional controls, machinery, and weight to a vehicle and propellers, and, if they are to be properly located for good hydrodynamic performance, often are severe impediments to cross-country operations unless complex propeller retraction gear is provided.

It is no surprise, therefore, that most Army "swimmers," which are designed primarily for cross-country operations, do not have any auxiliary propulsion device but rely wholly on what thrust they can obtain by simply spinning their wheels in the water. If the wheel is partially submerged, as in the GOER vehicles, the tire acts as a paddle wheel and moderate speeds (2-3 mph) may be obtained. If, on the other hand, the wheels are totally submerged, as in the XM656 the propulsive efficiency is still further decreased and only minimal ($1\frac{1}{2}$ to 2 mph) speeds are attainable.[1] Somewhat improved propulsion can be obtained by the use of suitable shrouding around the tires,[2] but those shrouding arrangements that substantially improve propulsion are totally unacceptable for cross-country operations.

There is, therefore, a need for a compatible propulsion system which will provide adequate thrust to yield reasonable water speeds, yet not interfere with the basic off-road mission of the vehicle. Such a concept, herein designated a "wheel pump," was conceived some time ago by the authors (Figure 1). Basically, the wheel-pump concept envisions some simple wheel alterations to enable the turning wheel to pump water axially toward the center of the vehicle into a simple, static device designed to redirect the flow rearward, thereby obtaining forward thrust.

This report describes the analysis, design, construction, and testing of a first-cut, study model to determine the feasibility and practicality of such a device. At this time, only a device pumping the water through the wheel disc between the wheel hub and the tire is considered.

ANALYSIS

This section presents a simplified general description of the basic flow phenomena using energy and momentum relations, interpreted in terms of empirical knowledge of the suitable operating regimes of alternate kinds of flow machines. The selection of suitable pump and turning vane characteristics for the present tests will then be described, citing references for detailed design procedures.

Insight into the principle of operation of the wheel pump can be obtained from consideration of simplified one-dimensional fluid dynamic relations for an actuator disk and turning system as illustrated in the sketch below:

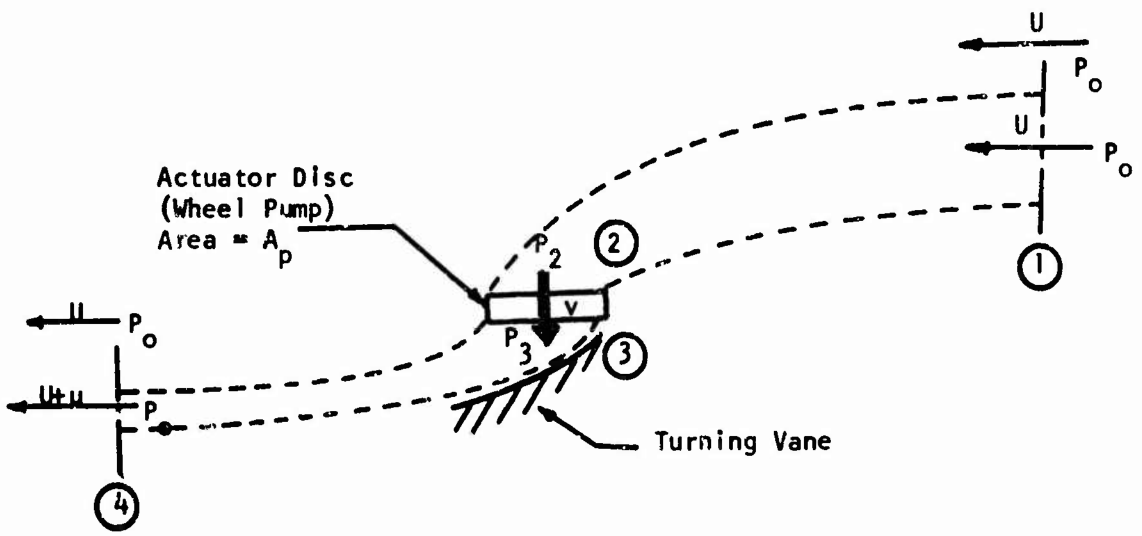

FIGURE 2. SIMPLIFIED PUMP SYSTEM SCHEMATIC

Several assumptions will be made for the sake of convenience:

1. No losses occur in the flow within the stream tube enclosed by the dotted lines, which retains its identity;
2. The flow through the pump is assumed to be entirely normal to the actuator disk so that the flow rate $Q = A_p v$; and
3. The turning vane produces a complete turning of the pump outflow into the direction of motion.

Admittedly, these assumptions may be far from the truth, especially for the present case of propelling amphibious vehicles, but the qualitative results of the analysis will be instructive.

Let the water enter the imaginary stream tube at velicity U, the velocity of the vehicle, and exit at velocity U + u due to the action of the pump. The thrust of the pump is generated by the momentum imparted to the exit flow:

$$T = \rho Q u = \rho A_p v u \quad . \tag{1}$$

The energy lost in the slipstream is residual kinetic energy left in the water:

$$E_{lost} = \frac{1}{2} \rho Q u^2 = \frac{1}{2} \rho A_p v u^2 \quad . \tag{2}$$

The efficiency of the system is the useful work (the thrust x velocity) divided by the sum of the useful work and the lost energy, or

$$\eta_{ideal} = \frac{TU}{TU + E} = \frac{2}{2 + u/U} \quad . \tag{3}$$

Applying Bernoulli's equation to the stream tube between Stations 1 and 2 and between 3 and 4 separately, since energy is added to the stream at the pump, it is possible to derive the pressure difference across the actuator disk:

$$p_3 - p_2 = \frac{1}{2}\rho U^2 \left[\left(\frac{u}{U}\right)^2 + 2\frac{u}{U} \right] . \tag{4}$$

Solving for u/U in terms of the pump thrust coefficient:

$$C_{T_p} = \frac{(p_3 - p_2)A_p}{\frac{1}{2}\rho U^2 A_p} ,$$

$$\frac{u}{U} = -1 + (1 + C_{T_p})^{\frac{1}{2}} , \tag{5}$$

and the ideal efficiency may be expressed as

$$\eta_{ideal} = \frac{2}{1 + (1 + C_{T_p})^{\frac{1}{2}}} \tag{6}$$

where, although the thrust on the actuator disk does not contribute directly to propelling the system, it does appear to control the "best" achievable system efficiency. The system thrust coefficient, non-dimensionalized on the basis of the actuator disk area and the uniform steam speed, may be obtained from Eqs. (1) and (5) as

$$C_T = \frac{T}{\frac{1}{2}\rho A_p U^2} = 2\left[-1 + (1 + C_{T_p})^{\frac{1}{2}}\right]\frac{Q}{A_p U} . \tag{7}$$

In pump design practice, two coefficients are used which specify the headrise and capacity of the pump in terms of the impeller's tip speed πnD. These are the pressure coefficient, ψ:

$$\psi = \frac{p_3 - p_2}{\frac{1}{2}\rho(\pi nD)^2} = C_{T_p}\lambda^2 \tag{8}$$

and the capacity coefficient, φ:

$$\varphi = \frac{Q}{A_p(\pi nD)} = \frac{Q}{A_p U}\lambda , \tag{9}$$

where λ is the advance coefficient:

$$\lambda = U/\pi n D \ . \qquad (10)$$

The thrust-producing system is to be designed to suit a particular drag coefficient for the vehicle in water, C_D, based on the vehicle's significant area A, seeking the most efficient solution possible. Eq. (7) may therefore be rewritten in terms of this drag coefficient and the pump parameters:

$$\frac{C_D A_S}{\Sigma A_p} = 2\,\frac{\varphi}{\lambda}\left[-1 + (1 + \psi/\lambda^2)^{\frac{1}{2}}\right] , \qquad (11)$$

where ΣA_p is the total flow area of all pump impellers in operation. Note that from Eq. (6) the thrust coefficient of the pump, C_{T_p}, should be as small as possible for good efficiency. Therefore from Eq. (8), the value of ψ/λ^2 should also be small.

Figure 3, taken from a paper by vanManen and Oosterveld[3], shows the relation between specific speed, σ, and specific diameter, δ. Best efficiency for various pumping machines of different geometric design lies within the cross-hatched area, where:

$$\sigma = n\left[\frac{Q^{\frac{1}{2}}\,2\pi^{\frac{1}{2}}}{(2gH)^{3/4}}\right] , \text{ and} \qquad (12)$$

$$\delta = D\left[\frac{(2gH)^{\frac{1}{4}}\,\pi^{\frac{1}{2}}}{2Q^{\frac{1}{2}}}\right] \qquad (13)$$

On this chart are plotted curves of constant pressure coefficient, ψ, and constant capacity coefficient φ.

For the M151 we may use the following vehicle characteristics:[4]

$C_D = 0.8$

$A_s = 11.28\ ft^2$

$A_p = .762$ per wheel

From this data and Eq. (11) we can now plot on Fig. 3 curves for several values of λ ($\lambda = 0.2$ corresponds to approximately 28 mph wheel speed and 3 mph water speed for the wheel pumps in the M151). If we desire to reduce the frictional losses experienced at the tire tread we should reduce the pump speed, thereby increasing λ. Therefore a radial (centrifugal) pump appears to hold the best promise.

Since the above analysis is quite simplified and neglects such important factors as losses due to viscous eddying in the pumps, collectors and turning vanes, it was decided to design and test two types of pumps, both of the mixed flow type: one having eight blades and the other having sixteen blades. Another pump, of the axial flow type was designed but was not tested. Detailed design procedures are contained in standard pump textbooks, such as that by Betz.[5] The flow collector and turning vanes were laid out to suit mechanical restrictions imposed by suspension and underbody arrangements.

FABRICATION

Employing the equations developed in the preceding section, wheel-pumps and collectors were designed and fabricated to fit the configuration of the M151 4x4 jeep. In order to avoid any structural modifications to the M151 suspension, wheel centerlines were moved outboard by 4 inches on each side. Figure 4 shows the two variations in impeller design (8 and 16 blades) tested. Figure 5 shows the 8-bladed impeller mounted on the rear wheel of the M151. Figures 6, 7 and 8 show the collectors mounted on the vehicle. It can be easily seen in Figure 7 that the collector imposes no steering restraints to the vehicle.

The wheel pump and collectors were both made of sheet steel. Upon completion, the combination impeller wheel and brake drum plus the collector weighed 46 pounds per wheel or 23 pounds more per wheel station than the standard wheel and brake drum it replaced. For actual service it would not be necessary to make the collector of steel. Some rubber-fabric material that would collapse when not in use, but inflate under water pressures during pumping, should be equally serviceable.

TESTS

The 8-blade and 16-blade wheel pumps were first mounted on a test stand (without tire) to obtain an estimate of their performance as pumps and then on the vehicle to evaluate their potential as a vehicle propulsor.

Pump Performance Tests

Performance of the wheel pump was evaluated prior to installation on the vehicle, using a simple recirculating tank test stand (Fig. 9). A schematic of this setup is shown in Fig. 10. Table I summarizes the results of these tests. Pump speed was measured by a tachometer. Input torque was measured by a damped spring-scale connected to the pivot-mounted engine. Flow velocity and head across the pump were measured by manometers and output horsepower was computed from

$$HP_o = \gamma \cdot v \cdot H \cdot A_v \tag{14}$$

where

γ = specific weight of water $(\mathrm{lb/ft})^3$
v = average water flow (ft/sec) at the measuring section
H = difference in head across pump (ft)
A_v = cross-sectional area where V was measured (sq ft)

TABLE I

RESULTS OF PUMP TESTS IN RECIRCULATED TANK TEST STAND

Test No.	Pump Speed (RPM)	Input Power (HP)	Velocity (FPS)	Input Head (FT)	Output Power (HP)	Efficiency	Number Blades	Throttle Position	Damper Position
2	212.6	1.71	2.21	2.6	0.08	0.048	8	Part	Open
3	248.8	2.67	2.62	3.7	0.13	0.052	8	Mid	Open
4	255.6	2.92	2.52	4.2	0.15	0.052	8	Full	Open
7	214.9	1.80	2.27	3.3	0.10	0.060	8	Part	Open
8	248.8	2.59	2.63	4.6	0.17	0.066	8	Mid	Open
9	255.6	2.83	2.59	4.7	0.17	0.061	8	Full	Open
12	217.2	1.75	1.91	5.1	0.13	0.079	8	Part	Part
13	251.1	2.70	2.36	7.2	0.24	0.090	8	Mid	Part
14	257.9	2.86	2.44	8.1	0.28	0.098	8	Full	Part
16	217.2	1.75	2.03	5.5	0.15	0.090	8	Part	Part
17	251.1	2.70	2.45	7.5	0.26	0.096	8	Mid	Part
18	257.9	2.86	2.21	8.0	0.25	0.088	8	Full	Part
21	210.4	1.48	0.65	17.3	0.16	0.108	8	Part	Closed
22	253.3	2.47	1.00	24.6	0.35	0.142	8	Mid	Closed
24	271.5	3.10	0.72	26.3	0.27	0.087	8	Full	Closed
26	217.2	1.60	0.79	18.5	0.20	0.129	8	Part	Closed
27	251.1	2.44	0.91	23.3	0.30	0.123	8	Mid	Closed
28	266.9	2.96	0.55	25.3	0.20	0.068	8	Full	Closed
31	217.2	1.82	1.84	1.1	0.02	0.016	16	Part	Open
32	248.8	2.76	2.00	1.2	0.03	0.012	16	Mid	Open
33	253.3	2.89	2.02	1.6	0.04	0.016	16	Full	Open
35	217.2	1.89	1.69	1.1	0.02	0.014	16	Part	Open
36	248.8	2.84	2.02	1.2	0.03	0.012	16	Mid	Open
37	248.8	2.84	2.22	1.6	0.05	0.018	16	Full	Open
39	217.2	1.89	2.07	2.5	0.07	0.038	16	Part	Part
40	239.8	2.57	2.44	3.1	0.10	0.042	16	Mid	Part
41	248.8	2.76	2.34	3.1	0.10	0.037	16	Full	Part
43	217.2	1.97	1.86	2.5	0.06	0.033	16	Part	Part
44	235.2	2.45	2.57	3.0	0.10	0.044	16	Mid	Part
45	248.8	2.84	2.48	3.2	0.11	0.040	16	Full	Part
47	217.2	1.67	0.56	18.2	0.14	0.087	16	Part	Closed
48	239.8	2.25	1.02	22.1	0.32	0.142	16	Mid	Closed
49	253.3	2.81	0.85	25.1	0.30	0.108	16	Full	Closed
51	217.2	1.67	0.85	17.7	0.21	0.128	16	Part	Closed
52	239.8	2.25	1.07	21.8	0.33	0.147	16	Mid	Closed
53	251.1	2.61	1.19	23.6	0.40	0.153	16	Full	Closed

The highest efficiency (15%) was obtained in Test No. 53 which was at maximum head rise.

Propulsion Performance Tests

To evaluate more realistically the performance of the wheel pump in a complete propulsion system, four pumps were mounted on the wheels of an M151 4x4 ¼-ton truck (with fording kit) supported in a raft in such a manner that the wheels could be operated at various axle depths below the still water surface (Fig. 11).[2] A load cell connecting the raft and the jeep was arranged to measure the horizontal fore-and-aft forces between the jeep and the raft (Fig. 12). The cables supporting the vehicle were kept vertical in the fore-and-aft centerline plane (Fig. 13). A schematic of this setup is shown in Fig. 14.

Two types of tests were run: Bollard pull tests, in which the raft was immobilized by a line to a piling and pull measured with no forward velocity; and free running tests which included tests in which the raft was towed by an auxiliary boat with and without the wheels and wheel-pumps running. Unless otherwise specified, the 16-blade pumps were installed on the rear wheels; the 8-blade pumps on the front.

Bollard Pull Tests

Results of the bollard pull tests of the four pumps mounted in the raft-supported M151 are presented in Figs. 15 to 37, and in Table II. Figure 15 summarizes the thrust obtained in bollard pull tests with the standard 7.50 x 16 NDCC military tires only (i.e., with no wheel pumps). The complete data are plotted in Figs. 16-19. Maximum thrust in all cases -- from all-wheel or rear (only) drive, with or without splash suppression skirts -- was of the order of 130 pounds.

Figure 20 similarly summarizes the static bollard performance of the wheel pumps, when the wheels were fitted with the same tires (7.50 x 16 NDCC); figures 21-24 show the detailed data. With the wheel pumps attached, maximum bollard pull is increased to 190-250 pounds, depending on drive configuration and skirt effects. The actual change in performance due to fitting the wheel pumps is shown in Fig. 25.

Figure 26 summarizes bollard pull tests (Figs. 27-30) in which smooth, treadless tires (6.50-16) were fitted in place of the military tires. Included in this test series were runs in which the 16-blade pumps installed on the rear wheels (Figs. 27 and 29) were interchanged with the 8-blade pumps (Figs. 25 and 30) normally fitted to the front wheels. The effect of the number of blades was minor, but the effect of the elimination of tire tread was appreciable. This reduction of power absorption by the tires, and, hence, the release of more power to the pumps, increased maximum thrust to the 320-370 pound range. The change in thrust characteristics resulting from the tire change is shown in Fig. 31.

As an indication of the power absorbed by the tire tread, tests were made at wide open throttle on an M151 in the water channel at the Land Locomotion Laboratory. Table II shows the results of these tests along with estimated delivered engine power calculated from the published power train characteristics also shown in Table II.

Tests in which the stationary reaction collectors were crudely altered to decrease their outlet area by 50 percent (Fig. 32) are summarized in Fig. 33; backup data are in Figs. 34-36. In this test series, in addition to testing four-wheel drive and rear-wheel drive (only), front-wheel (only) drive was tested. Figure 37 shows that, in the bollard pull tests at least, the front and rear wheel pumps behaved independently, with the rear pumps and tires performing with some 30 percent greater efficiency. In Fig. 33, performances with full and 50 percent outlet areas are compared. The results indicate that the original outlet area (approximately 80 sq. in.) was near optimum for the designed power loading from the viewpoint of bollard pull.

Maximum thrusts recorded in these tests occurred at different wheel/wheel-pump speeds (here recorded as indicated speedometer speed) and in several gears. They thus also correspond to somewhat different net power available to the wheels. An approximate correction was made for this by estimating the nominal gross power available for each test from the published power train characteristics.[6] The results of these calculations are summarized in Table III in which pounds of static bollard pull/gross horsepower are shown for each maximum pull developed during the test series.

TABLE II

TESTS ON M151 WITH SUBMERGED WHEELS

Drive	Gear	Maximum Speedometer Speed (mph)	Equivalent Wheel Speed (rpm)	Equivalent Engine Speed (rpm)	Estimated Delivered Power (hp)
2-wheel	1	18	208	6150	11
	2	28	324	5340	32
	3	28	324	2800	56
	4	24	278	1440	32
4-wheel	1	18	208	6150	11
	2	20	232	3820	61
	3	20	232	2100	45
	4	15	174	900	20

PUBLISHED POWER TRAIN CHARACTERISTICS[6]

Engine: 141.5 cu. in.
71 hp @ 4000 rpm
44 hp @ 1800 rpm

Tire: 7.50 x 16 NDCC (665 rev/mi.)

Transmission ratios: 1.00 1.674, 3.179, 5.712:1

Transfer ratio: 1:1

Axle ratio: 4.86:1

TABLE III

BOLLARD PULL - POUNDS/HORSEPOWER

Tires	Pumps		Skirts	50% Nozzle	Axle Depth								
					12"			16"			20"		
					Pumps Driven								
	Front	Rear			Four	Rear	Front	Four	Rear	Front	Four	Rear	Front
6.50-16, SMOOTH	8	16	None	No	5.[illegible]	4.9	-	-	-	-	4.8	4.4	-
	16	8	None	No	5.4	5.5	-	-	-	-	-	-	-
7.00-16, NDCC	8	16	None	No	3.2	3.2	-	2.5	3.0	-	-	-	-
	8	16	Part	No	3.8	3.3	-	2.7	3.3	-	-	-	-
	8	16	Full	No	3.4	3.3	-	-	-	-	-	-	-
	8	16	Full	Yes	3.4	3.0	2.6	2.9	2.7	1.6	2.6	2.6	1.3
	None		None	No	1.9	1.9	-	1.1	2.0	-	1.4	1.9	-

Free Running Tests

The M151 mounted in the raft was towed at various speeds by an auxiliary boat as shown in Figure 14. The compression reading on the load cell during towing indicated the force required to propel the vehicle at the measured speeds hence the drag of the vehicle. Figure 38 shows a plot of load cell push vs measured water speed (curve A). When the vehicle was allowed to propel itself, the load cell exerted a force on the raft (its drag) which was measured and also plotted on Figure 38 as curve B. The sum of the two (curve C) indicates the combined drag of both the jeep and the raft and also indicates the approximate thrust the vehicle was generating at each measured speed. Extrapolating curve C to obtain the thrust generated at the maximum obtained speed (2.7 mph) yields approximately 135 lbs of thrust. This value projected on an extrapolation of curve A yields a predicted vehicle (only) speed of 3.2 mph, which is close to the 3.0 mph measured when the vehicle/raft system was being towed by the boat; the jeep was operated at wide open throttle; and the load cell measured no force between the raft and the vehicle.*

Table IV is a presentation of the maximum speeds achieved at free running condition (jeep and raft) with the vehicle powered, not towed by the boat.

During the free running tests it was apparent that increased vehicle control was obtained with the wheel pumps. Although not quantitatively measured, the turning radius was materially reduced and the response to steering input greatly improved.

*Tests conducted later (October 1969) at Houghton, Michigan with the wheel pumps mounted on a floating version of the M151 yielded a maximum vehicle speed of 3.2 mph.

TABLE IV

SPEED TESTS - SELF-PROPELLED

Tires	Center Blades		Skirts	Nozzle	Axle Depth					
					12" Pumps Driven		16" Pumps Driven		20" Pumps Driven	
	Front	Rear			Four	Rear	Four	Rear	Four	Rear
Smooth	8	16	Yes	No	2.6	-	-	-	2.6	-
	16	8	Yes	No	2.5	2.7	-	-	2.6	-
NDCC	8	16	Part	No	2.4	2.0	2.3	2.1	-	-
			Full	Yes	2.4	2.3	-	-	-	-

SUMMARY OF RESULTS

The data obtained in the test program indicates that the maximum bollard pull obtained using the wheel pumps with standard tires (230 pounds), is less than the 350 pounds predicted at the start of this program. The maximum thrust obtained by the pumps when operating with smooth tires (370 pounds) is closer. For standard tires alone, the maximum bollard pull was 134 pounds.

Test reports by Ford Motor Company indicates that a M151 with a simple flotation hull will travel 2.2 mph when propelled by its tires only. On the present test rig, with military tires and the wheel pump, the maximum free running speed (with the raft attached) was 2.4 mph; with smooth tires, the rig could obtain 2.7 mph; with military tires and the nozzle attached, it could obtain 2.4 mph.

While towing the rig and powering the jeep, the load cell read near zero when proceeding at 3.0 mph.

The wheel pumps materially improved vehicle control while afloat.

CONCLUSIONS

The results of the test program indicate that the wheel pump is able to generate considerable thrust, though not as much as originally predicted.

The parasitic drag of the tire treads seriously degrades the performance of the pump.

The high hydrodynamic drag of the M151 and the steep slope of the resistance curve (approximately at the 2.5 power) indicate that large increases in output by use of more power or improved efficiency will be required before this device will propel the M151, as presently designed, at a speed much higher than 3 mph. Alternatively, the hull of the vehicle must be redesigned to improve its drag characteristics.

The present design was optimized for maximum bollard pull (thrust at zero speed). Optimizing the pump for maximum speed may enable it to travel at only a marginally higher speed in view of the statements contained in the preceding paragraph.

The improved control generated by the wheel pumps may be more important a factor than the marginally improved speed, since a major problem of wheel-propelled floating vehicles is steering control.

RECOMMENDATIONS

1. That the wheel pumps be mounted on a standard military vehicle built to operate afloat (such as the M561 or the M656) to determine its operational characteristics.

2. That a design effort be initiated which would enable the vehicle to take advantage of the power absorbed by the tire treads. A sketch of such a concept is shown in Fig. 39.

3. That further design studies on the collector be conducted to yield greater thrust in the vicinity of 3 mph.

ACKNOWLEDGEMENTS

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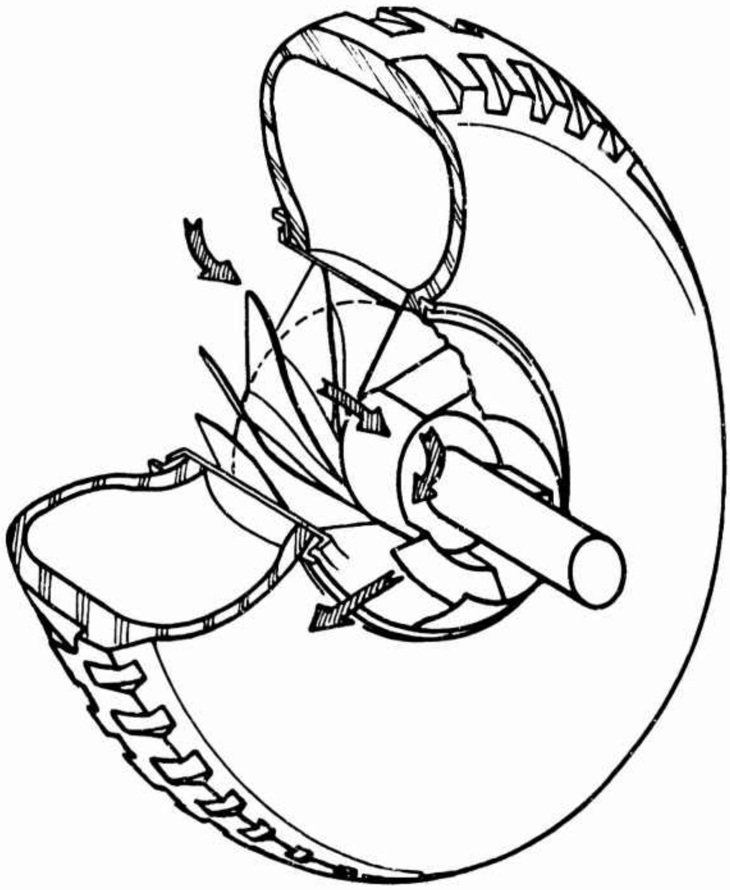

FIGURE 1. EARLY WHEEL PUMP CONCEPT SKETCH

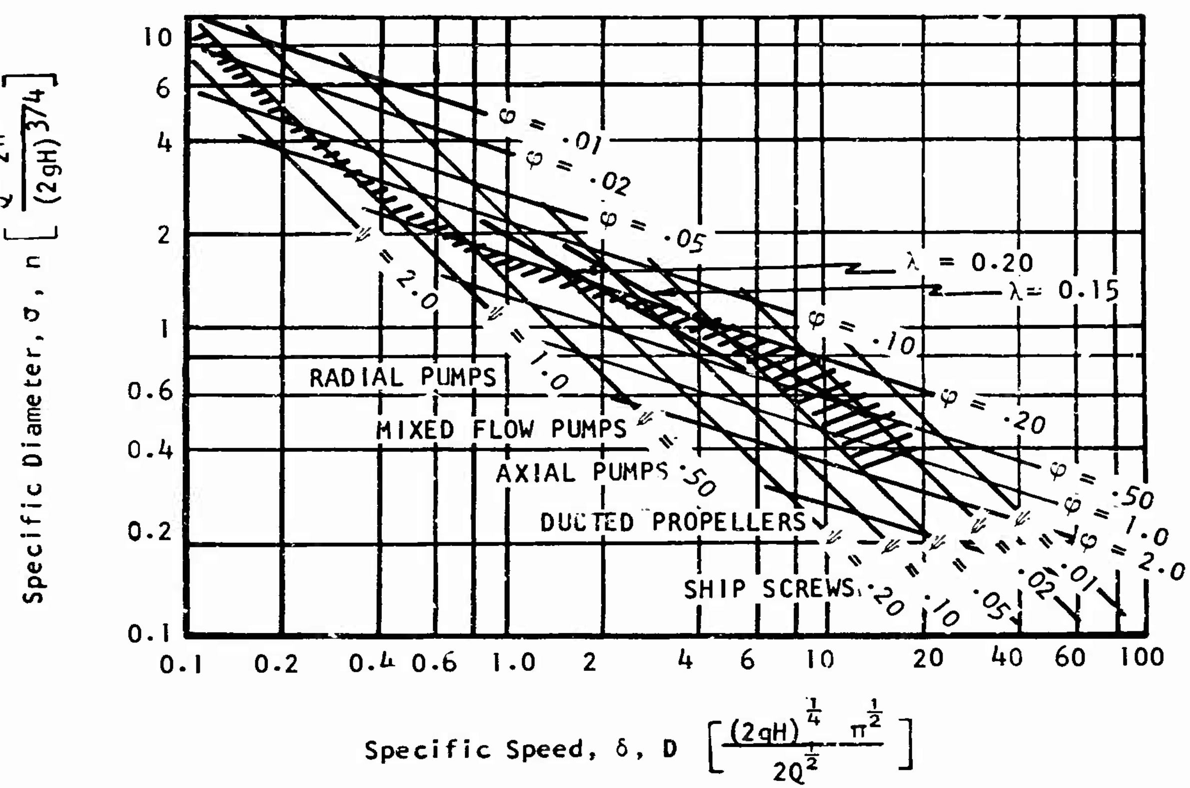

FIGURE 3. RELATION BETWEEN SPECIFIC SPEED, δ, SPECIFIC DIAMETER, σ, FOR VARIOUS PRESSURE COEFFICIENTS, ψ, AND CAPACITY COEFFICIENTS, φ. THE CROSS-HATCHED AREA INDICATES REGIONS OF BEST EFFICIENCY.[3] TWO VALUES FOR ADVANCE COEFFICIENT, λ, ARE ALSO PLOTTED ON THE CURVE

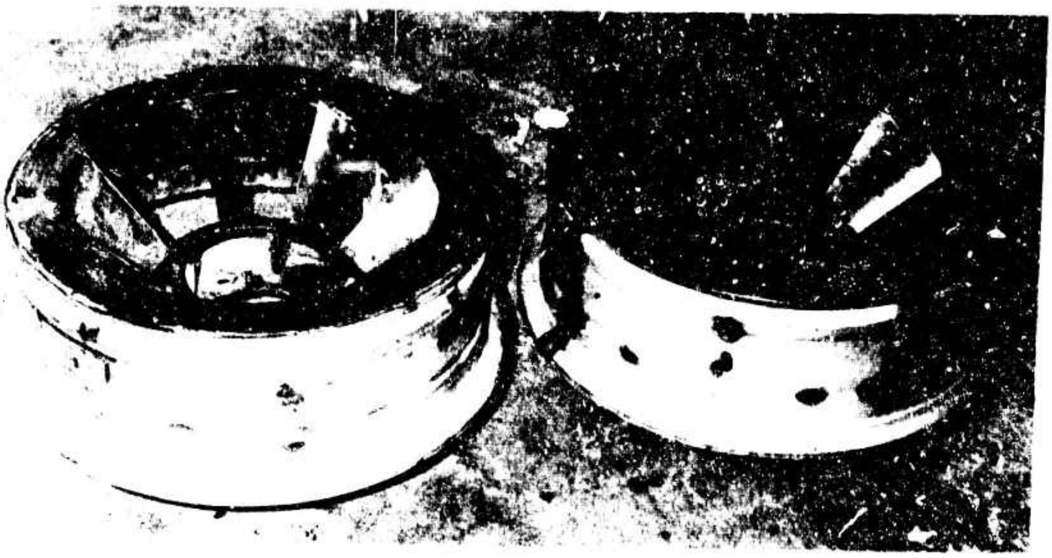

FIGURE 4. EIGHT- AND SIXTEEN-BLADED PUMPS EMPLOYED DURING THE PROGRAM

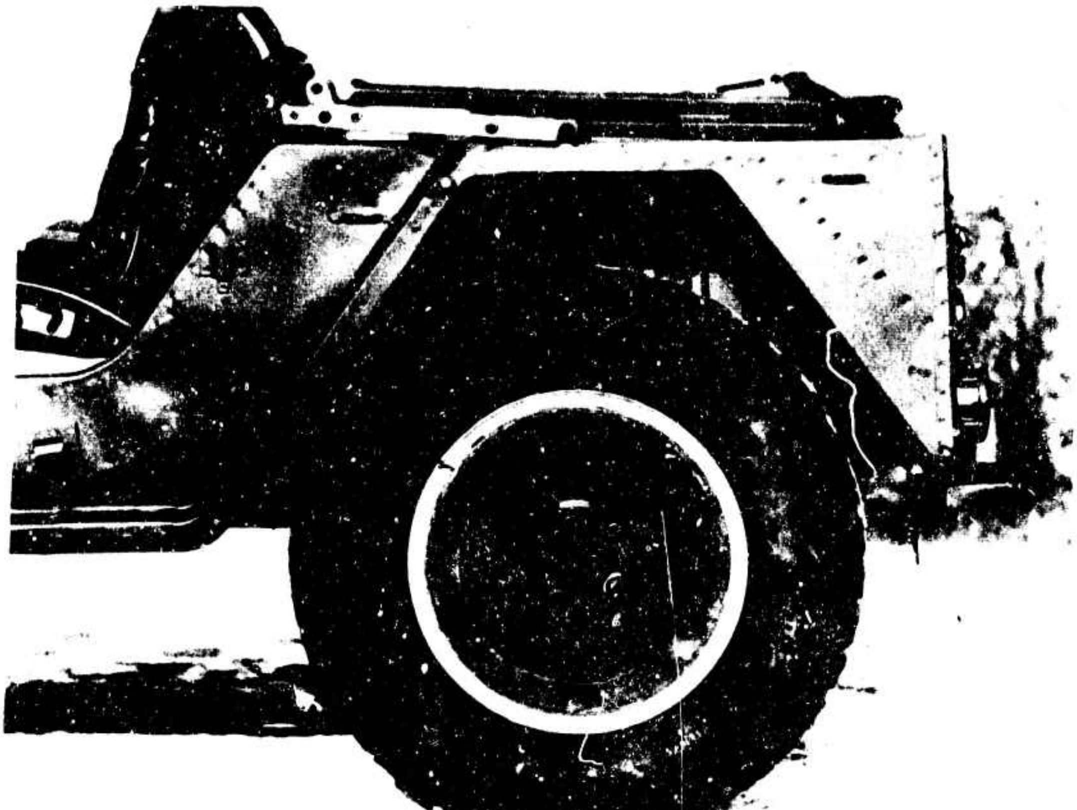

FIGURE 5. AN EIGHT-BLADED PUMP MOUNTED ON THE M151 $\frac{1}{4}$-TON TEST VEHICLE

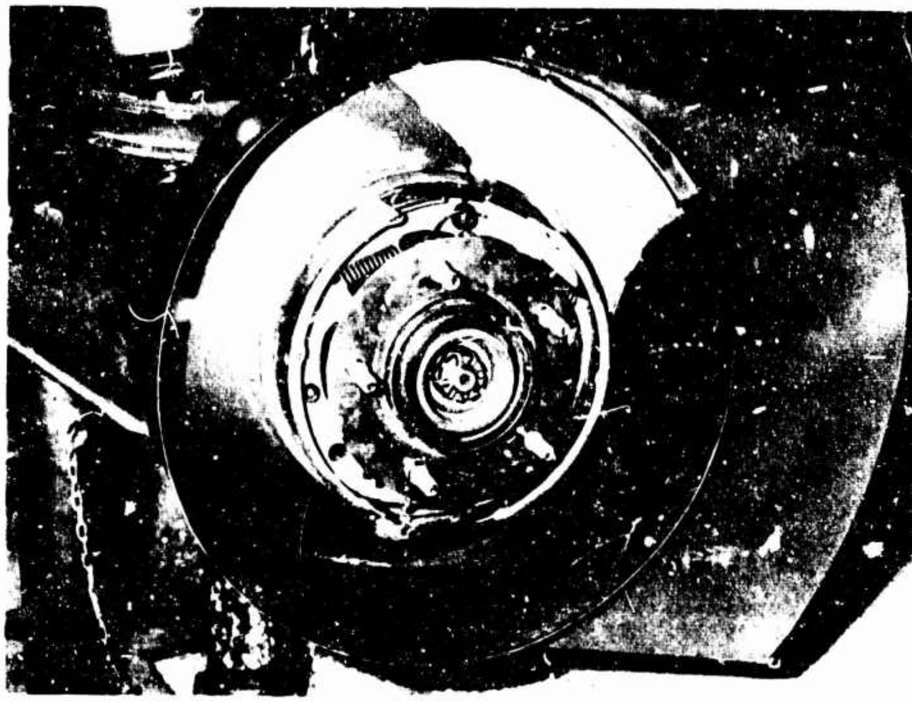

FIGURE 6. WATER COLLECTOR MOUNTED ON VEHICLE (SIDE VIEW)

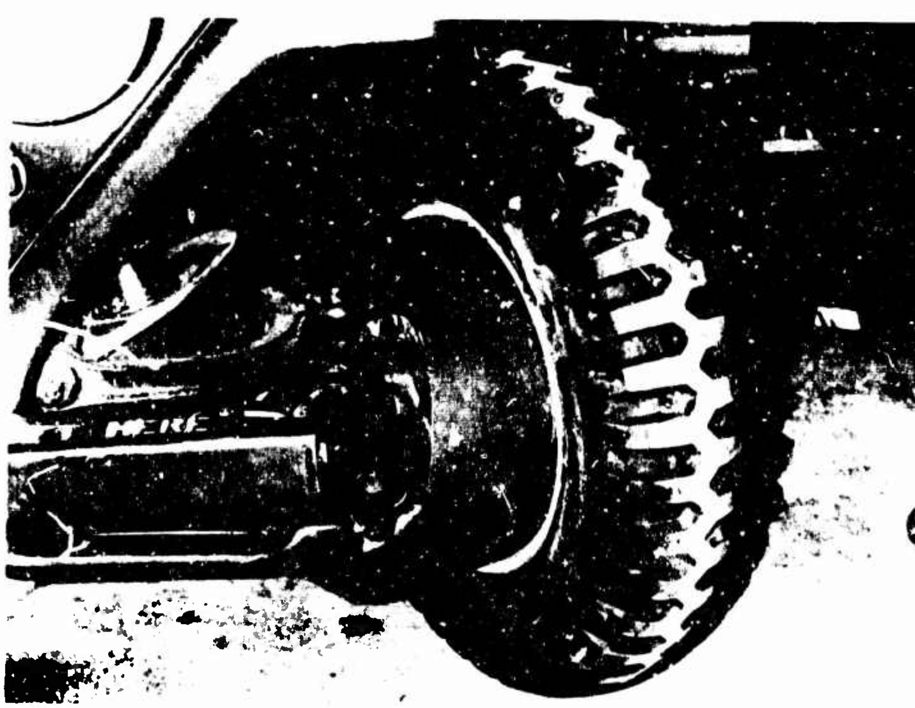

FIGURE 7. WATER COLLECTOR MOUNTED
ON FRONT SUSPENSION (FRONT VIEW)

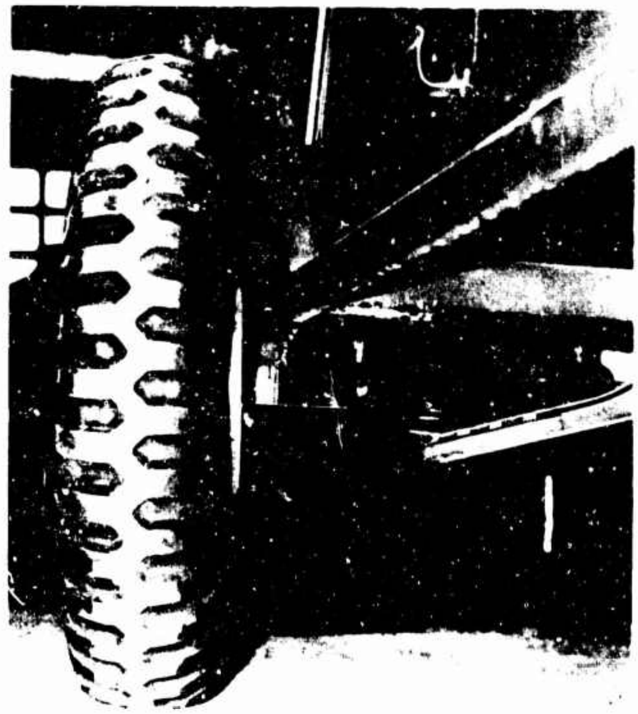

FIGURE 8. WATER COLLECTOR MOUNTED ON FRONT SUSPENSION (REAR VIEW)

FIGURE 9. RECIRCULATING TANK TEST STAND USED TO M ASURE PUMP OUTPUT AND EFFICIENCY

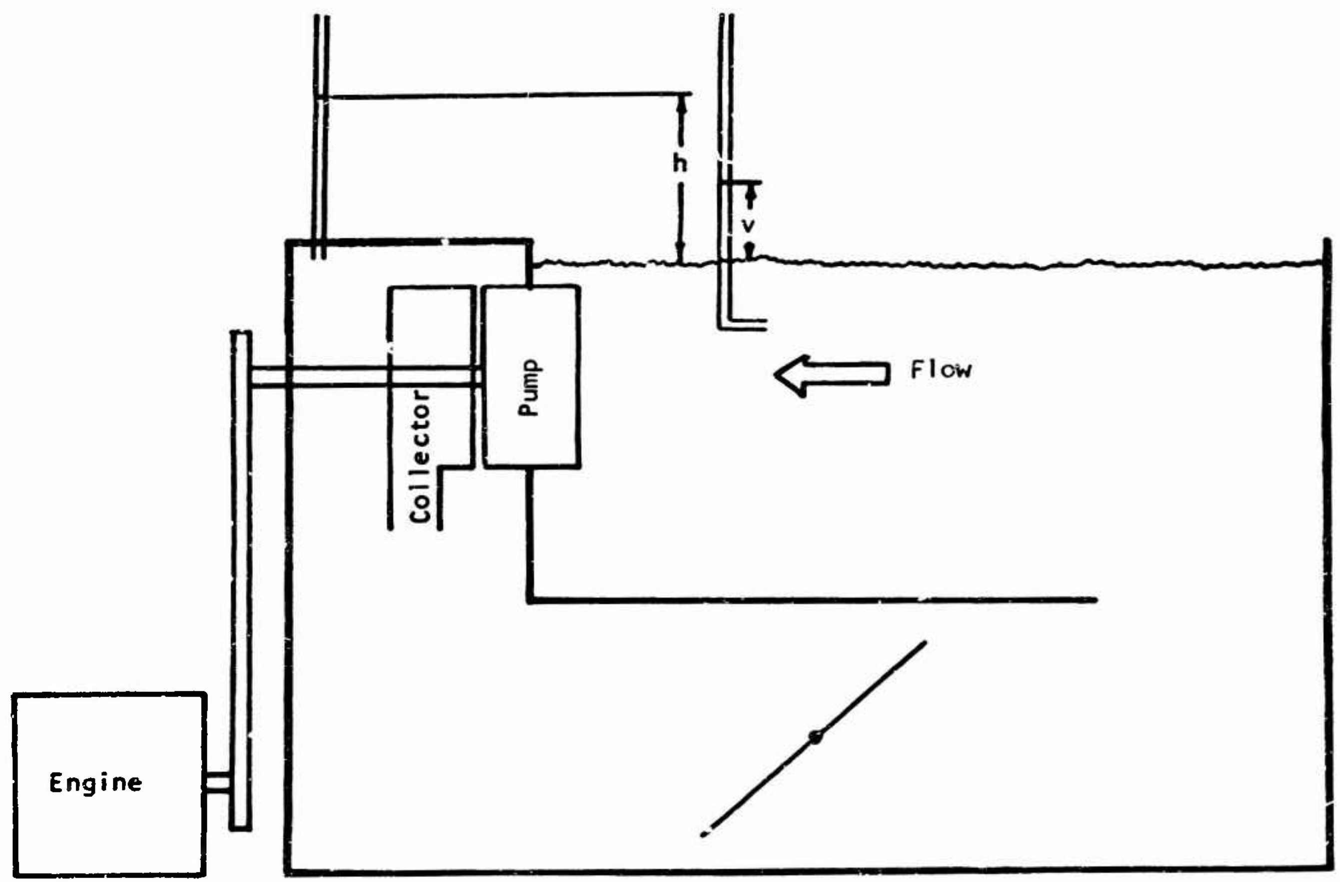

FIGURE 10. SCHEMATIC DRAWING OF THE RECIRCULATING TEST STAND

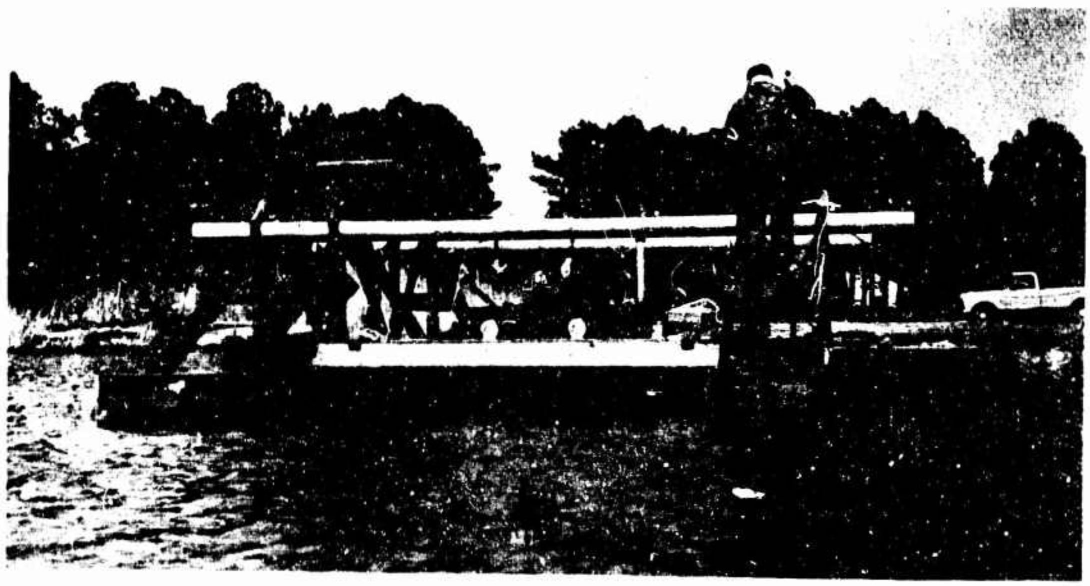

FIGURE 11. TEST VEHICLE MOUNTED IN SUPPORT RAFT

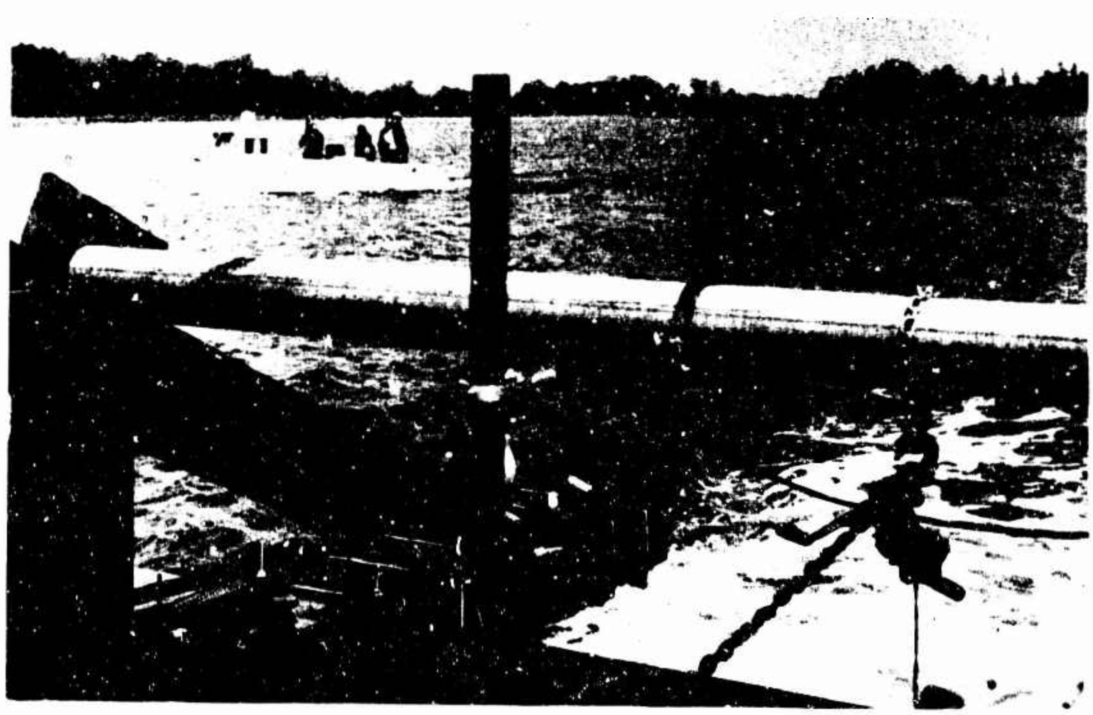

FIGURE 12. LOAD-CELL CONNECTION BETWEEN TEST VEHICLE AND SUPPORT RAFT

FIGURE 13. VEHICLE DURING OPERATION, SHOWING SUPPORT CABLES

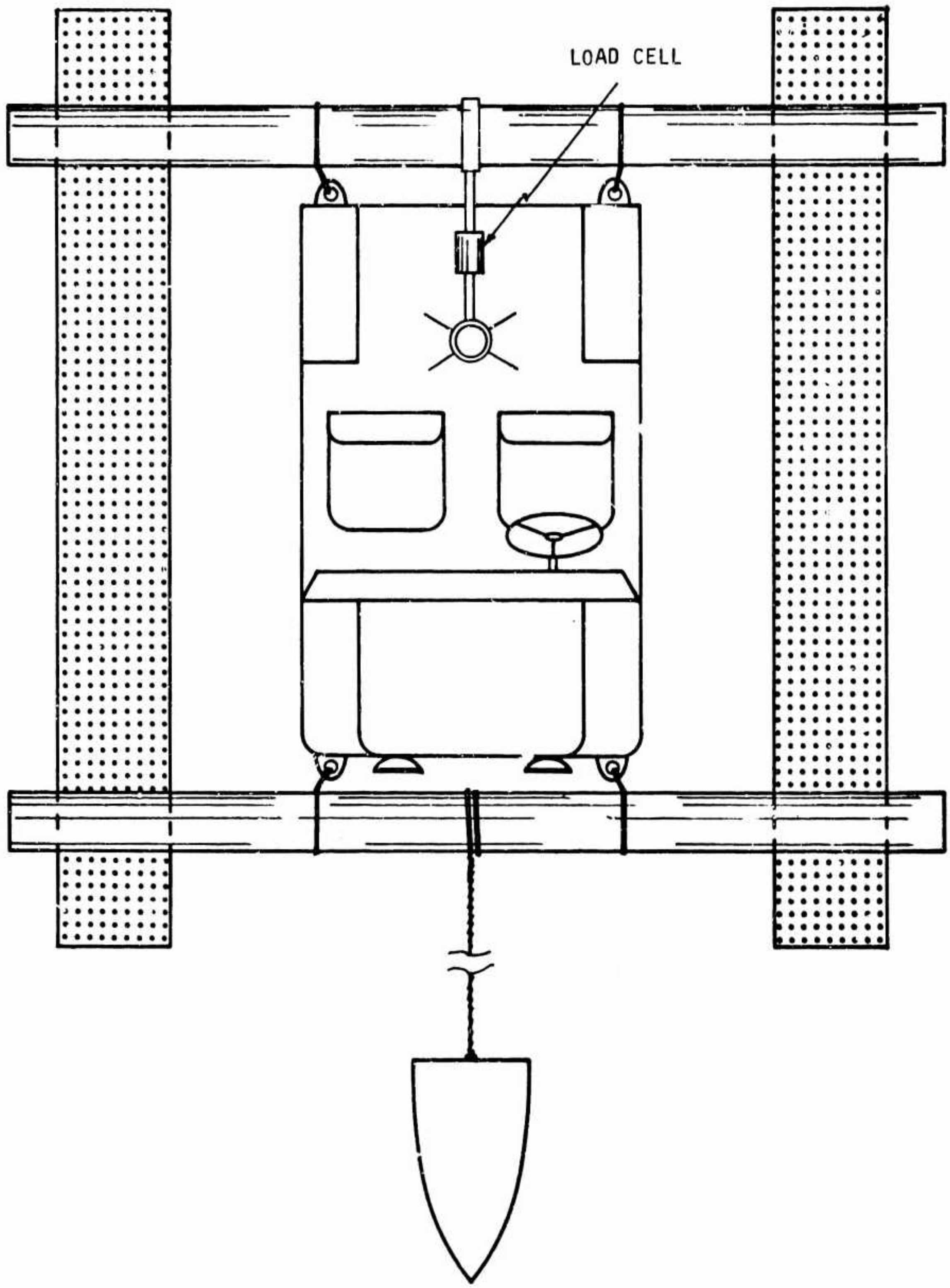

FIGURE 14. SCHEMATIC OF TEST VEHICLE/SUPPORT RAFT ARRANGEMENT WHEN TOWED BY BOAT

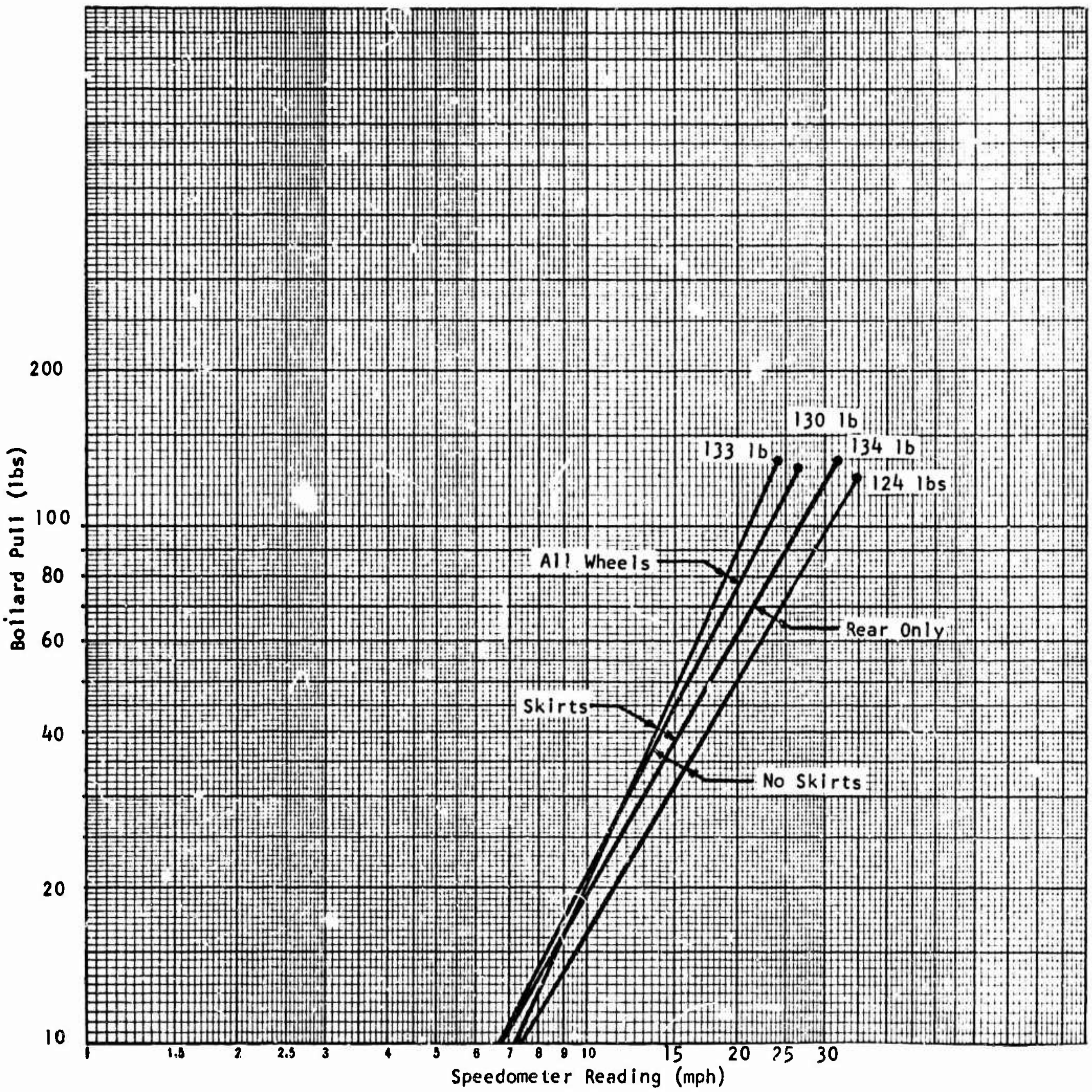

FIGURE 15. SUMMARY OF BOLLARD PULL TESTS - TIRES ONLY WITHOUT WHEEL PUMPS FROM FIGURES 16-19

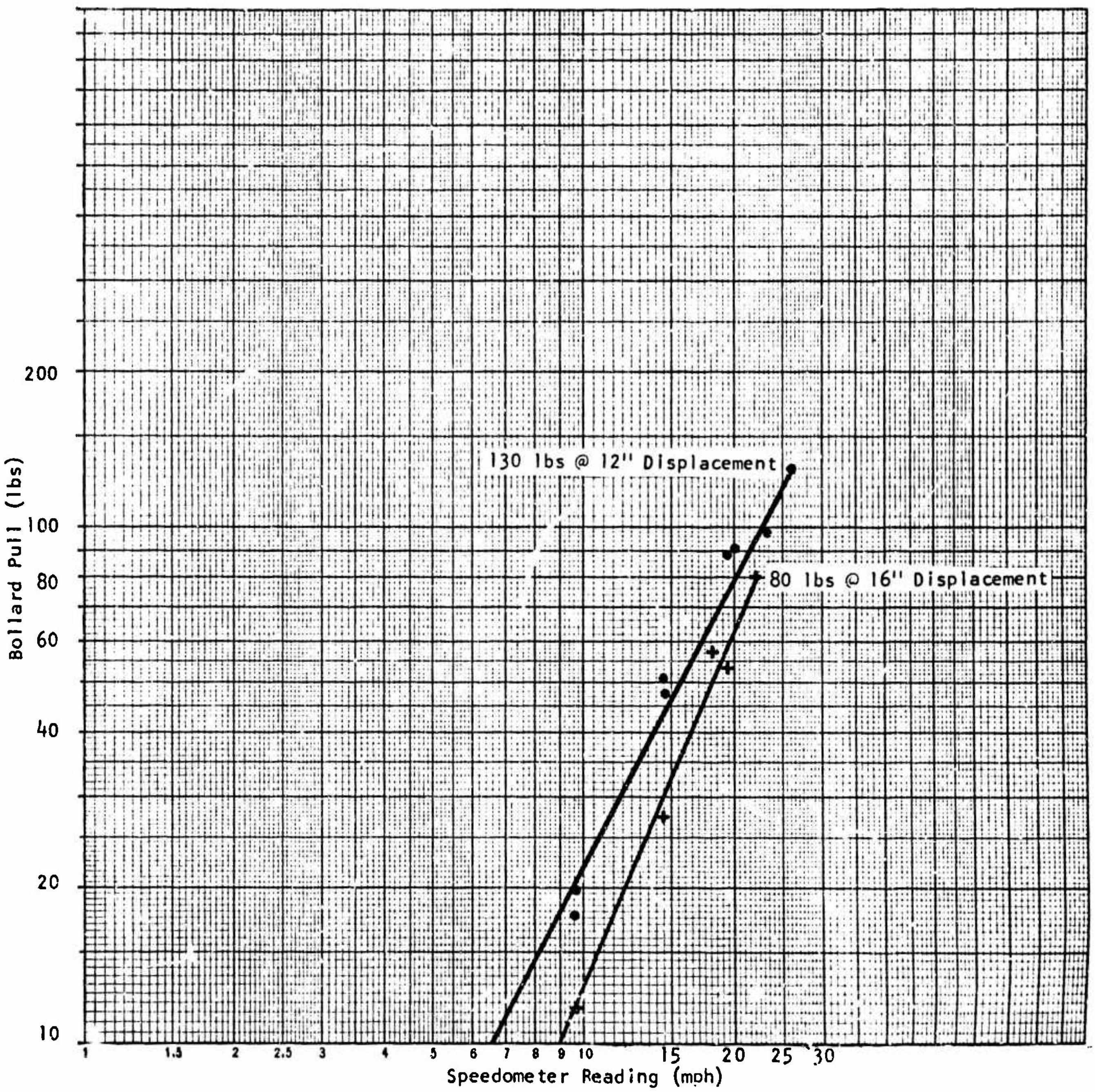

FIGURE 16. BOLLARD PULL TESTS - FOUR WHEEL DRIVE, NO WHEEL PUMPS, 7.50-16 NDCC TIRES, NO SKIRTS

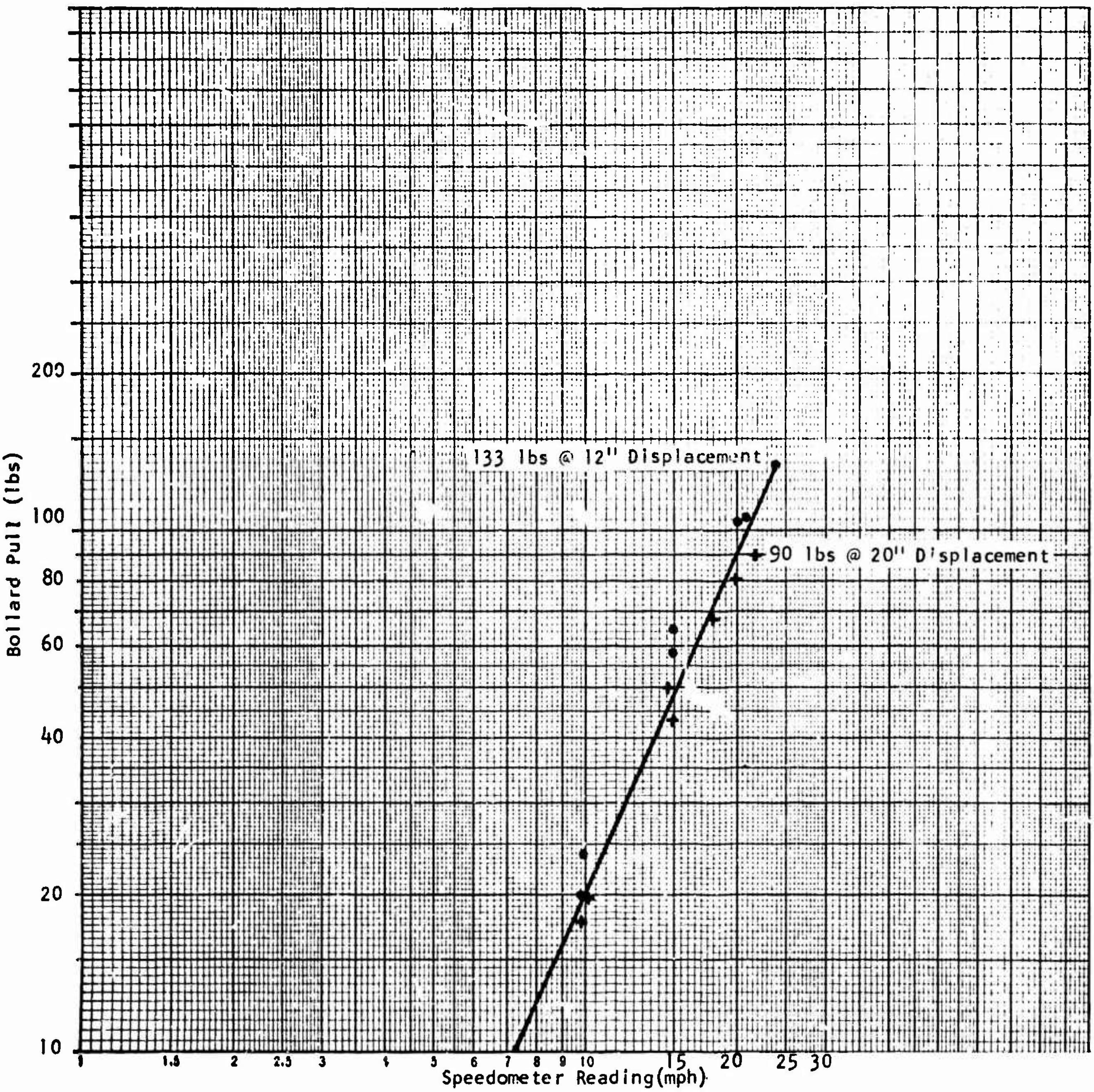

FIGURE 17. BOLLARD PULL TESTS - FOUR WHEEL DRIVE, NO WHEEL PUMPS, 7.50-16 NDCC TIRES, SKIRTS

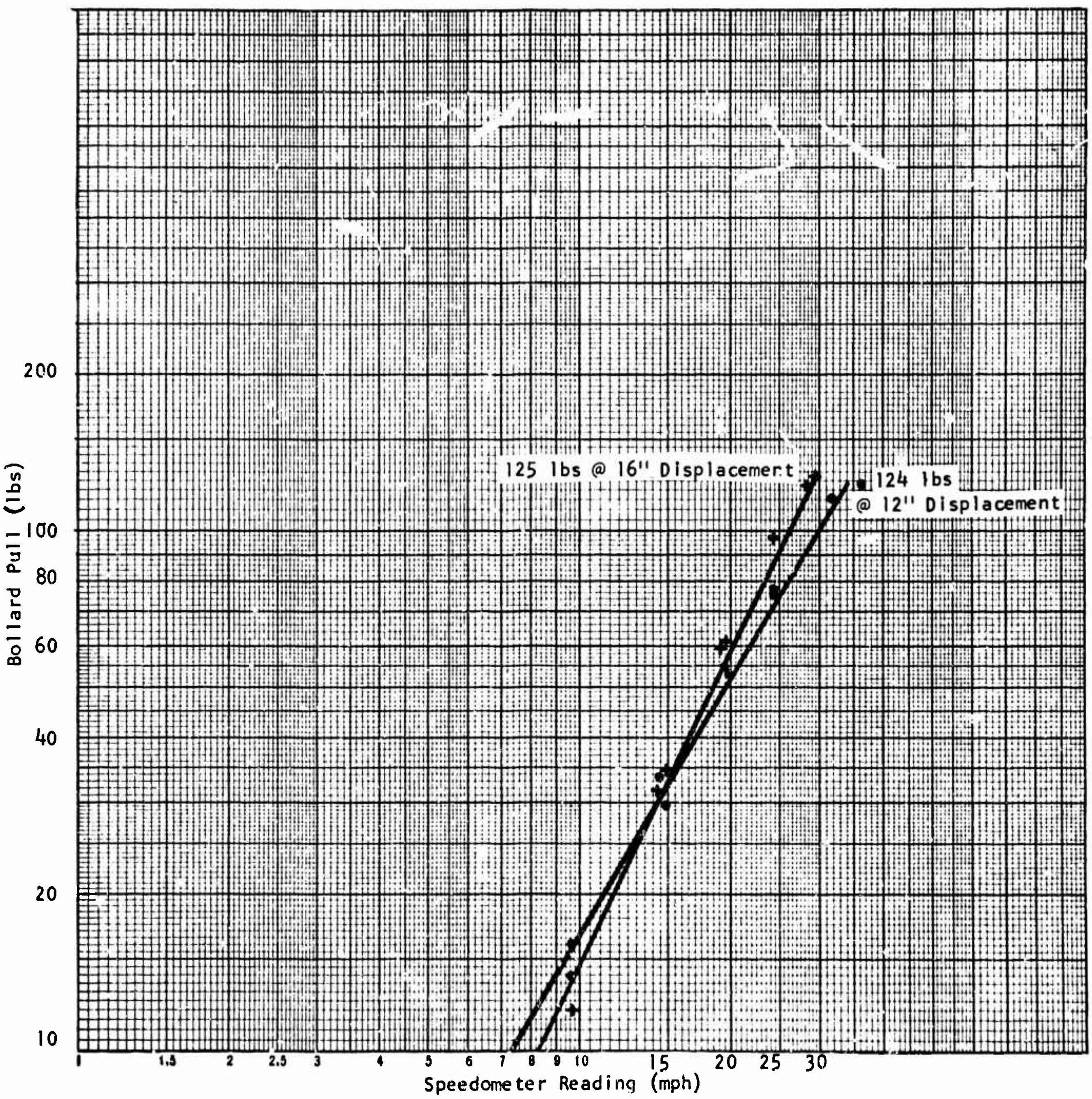

FIGURE 18. BOLLARD PULL TESTS - REAR WHEELS ONLY, NO WHEEL PUMPS, 7.50-16 NDCC TIRES, NO SKIRTS

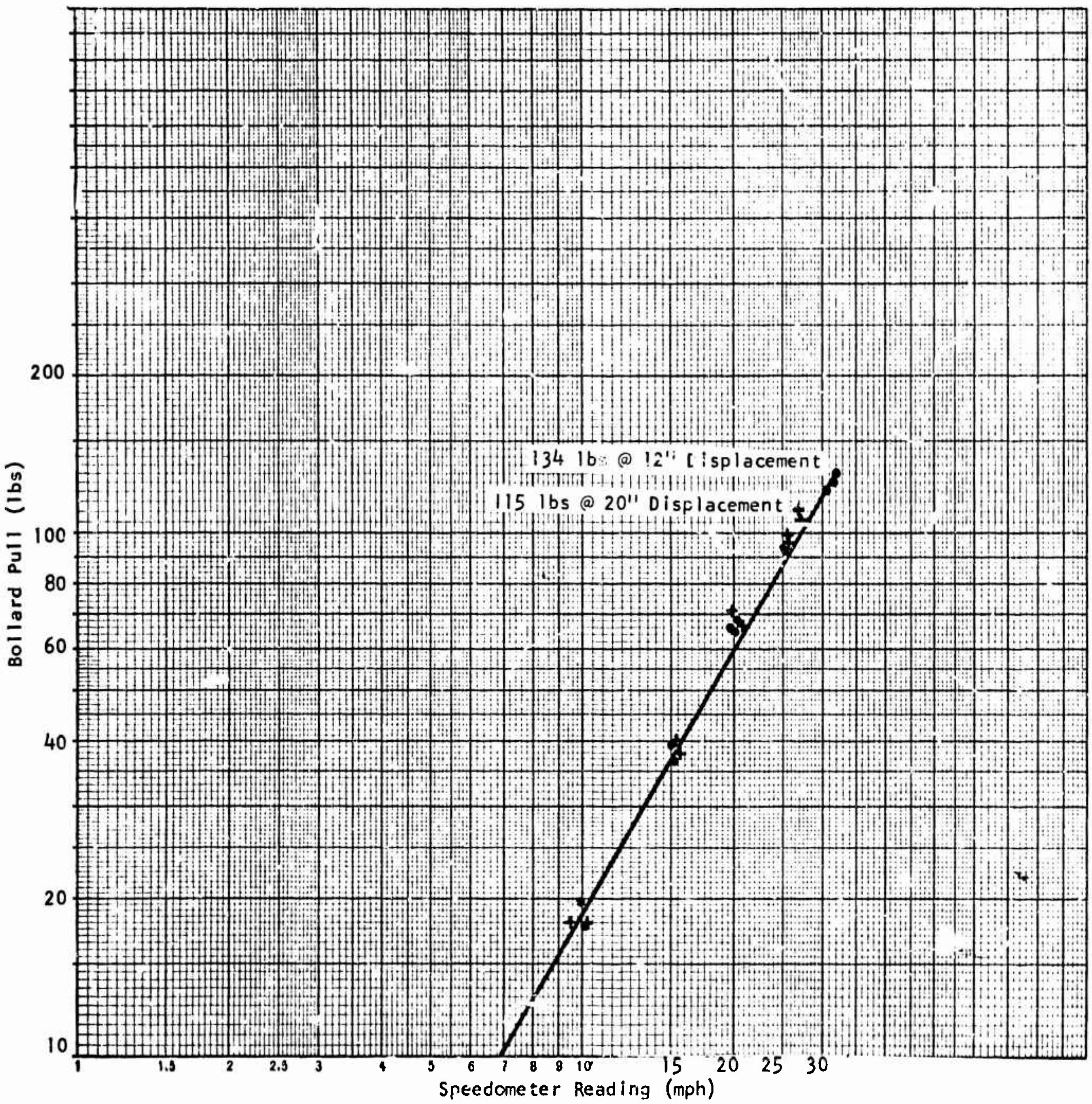

FIGURE 19. BOLLARD PULL TESTS - REAR WHEELS ONLY, NO WHEEL PUMPS, 7.50-16 NDCC TIRES, SKIRTS

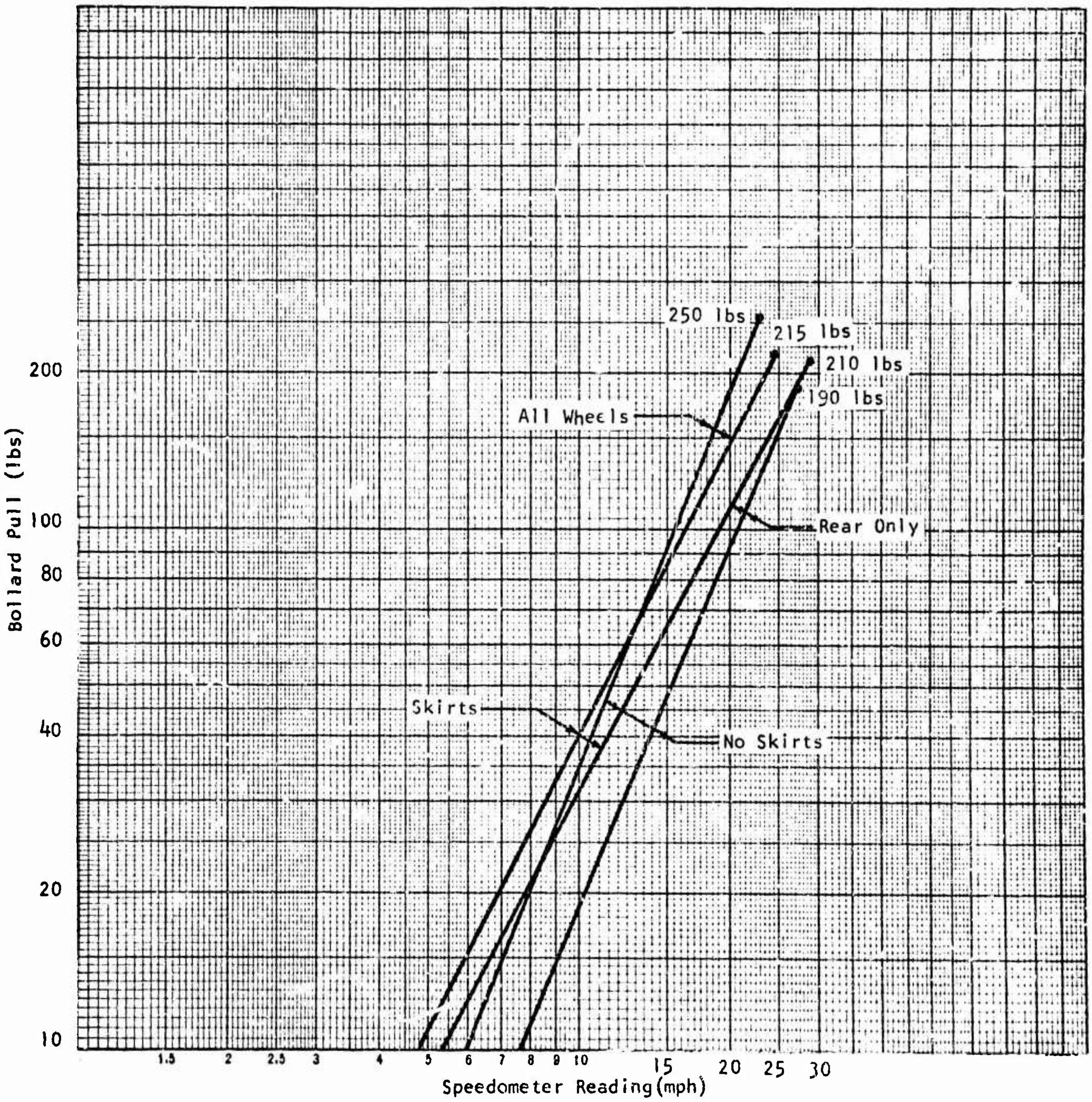

FIGURE 20. SUMMARY OF BOLLARD PULL TESTS - TIRES WITH WHEEL PUMPS, FROM FIGS. 21-24

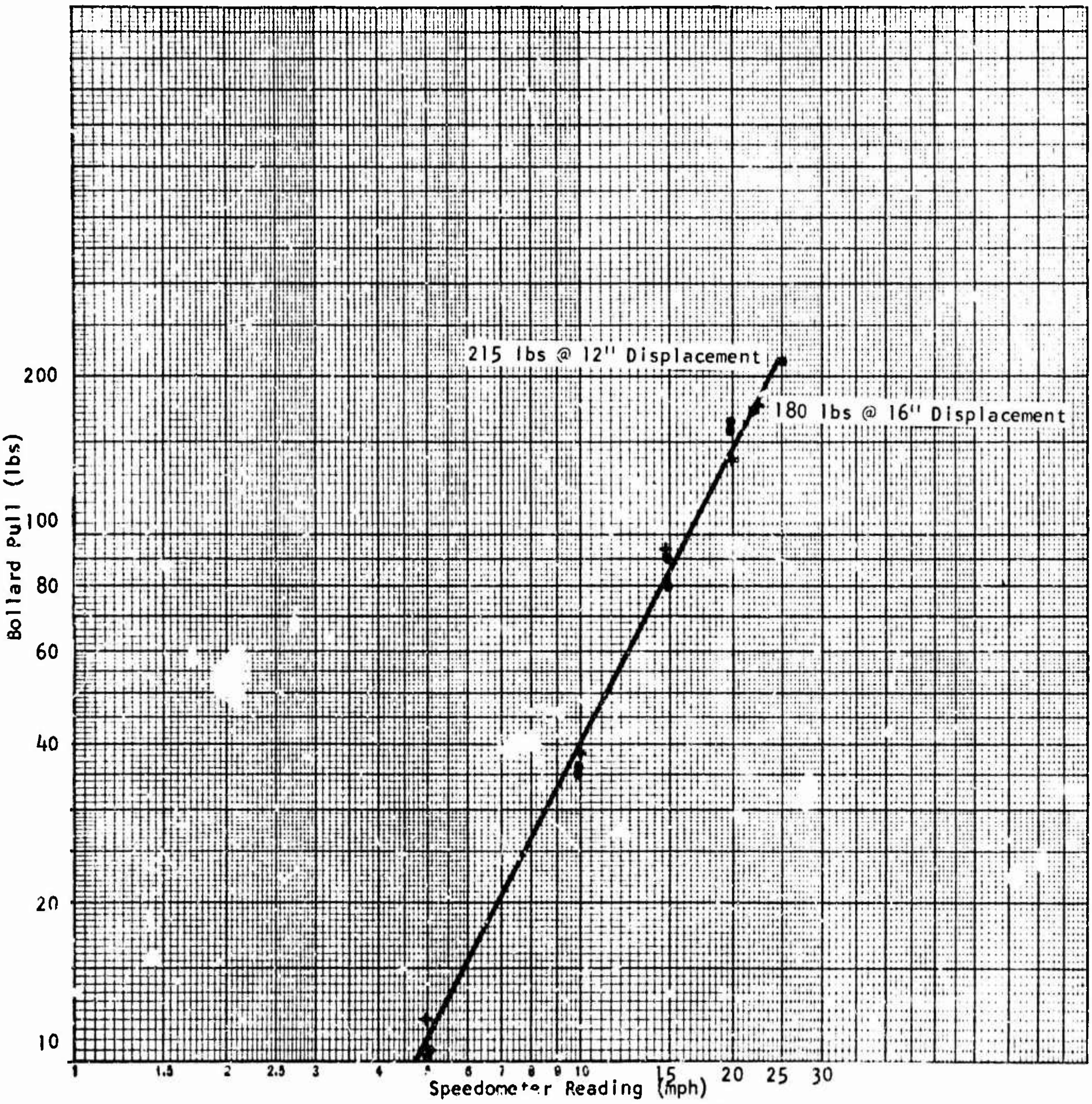

FIGURE 21. BOLLARD PULL TESTS - FOUR WHEEL DRIVE, WHEEL PUMPS, 7.50-16 NDCC TIRES, NO SKIRTS

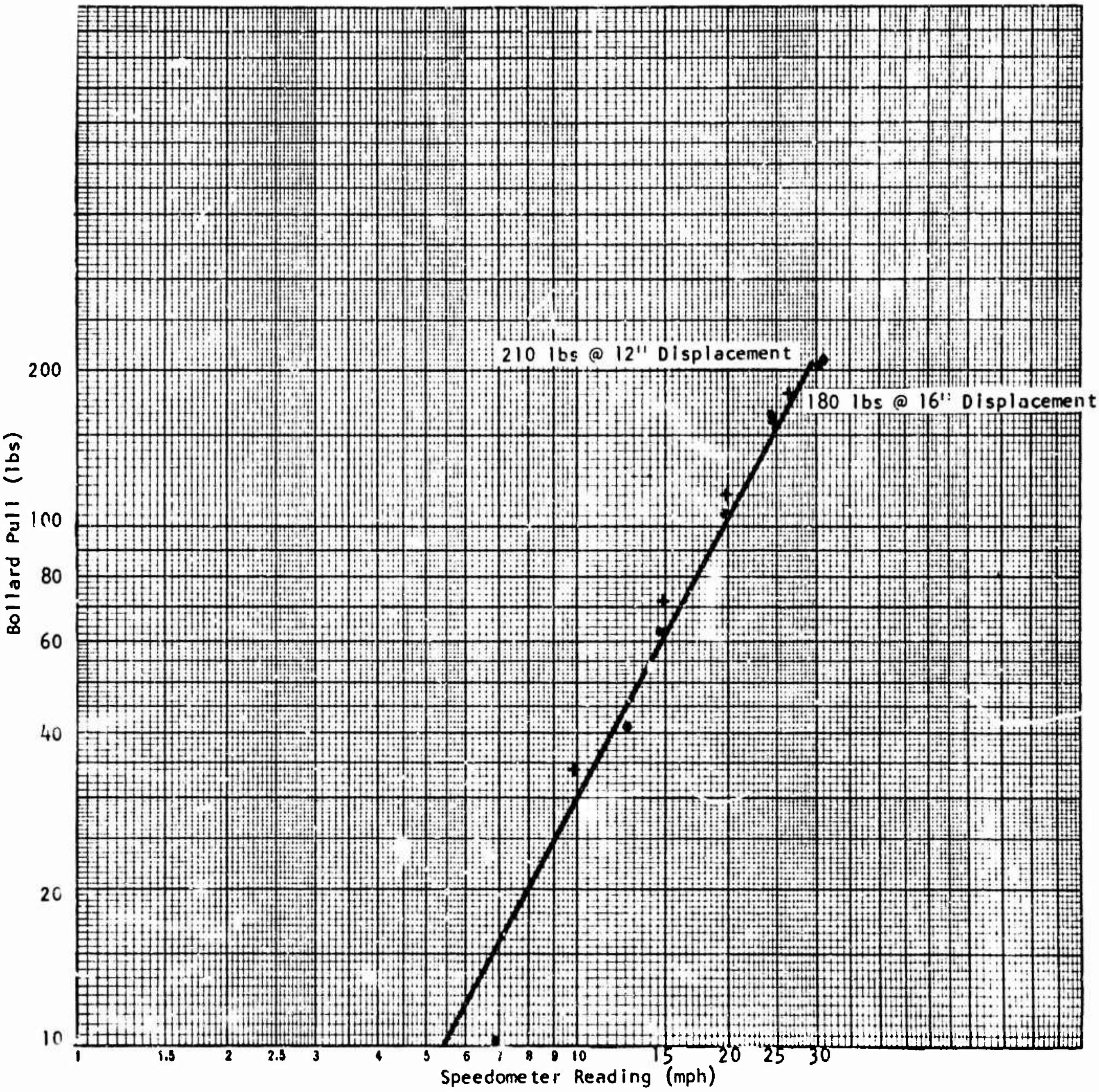

FIGURE 22. BOLLARD PULL TESTS - REAR WHEELS ONLY, WHEEL PUMPS, 7.50-16 TIRES, NO SKIRTS

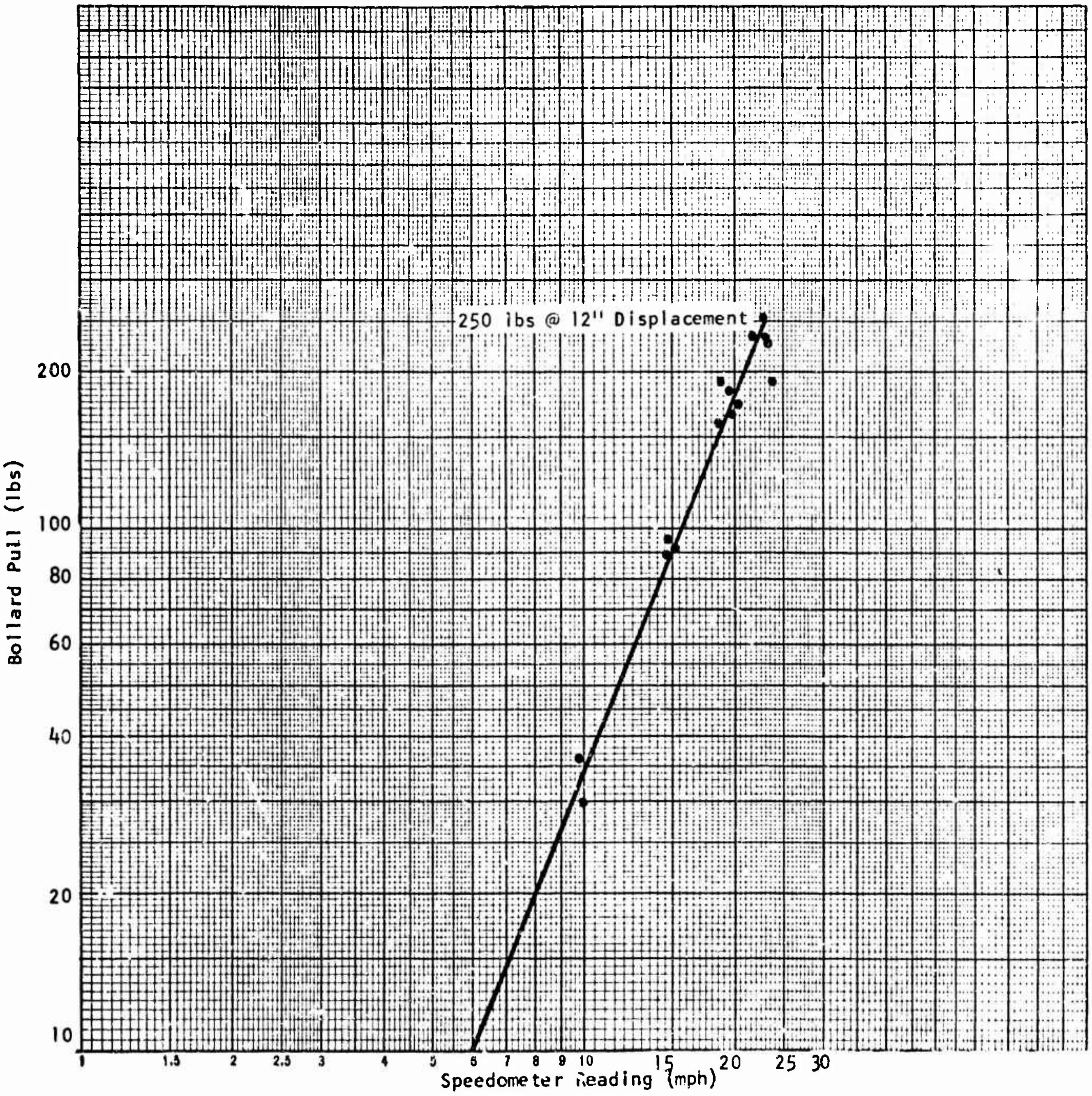

FIGURE 23. BOLLARD PULL TESTS - FOUR WHEEL DRIVE, WHEEL PUMPS, 7.50-16 TIRES, SKIRTS

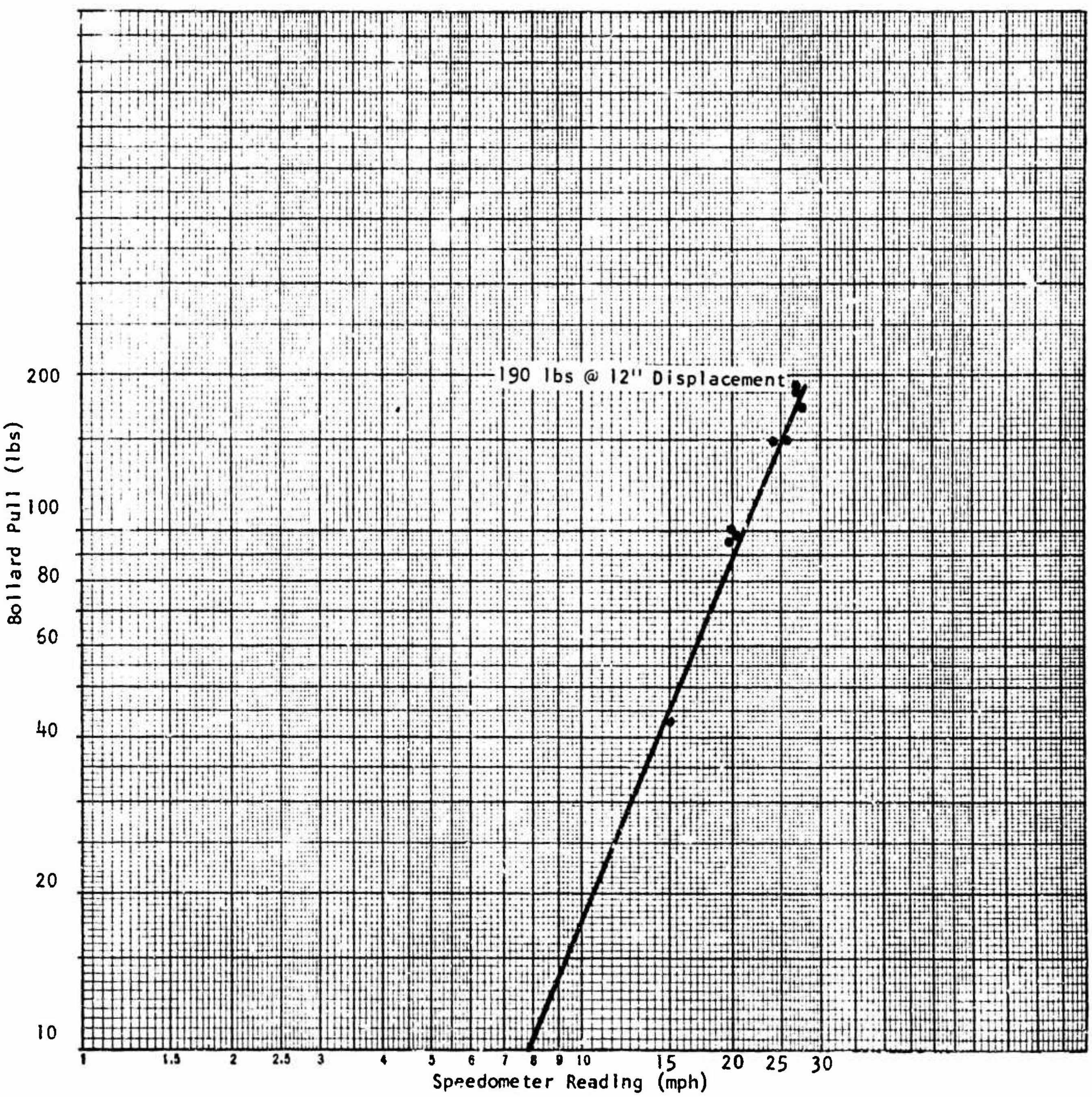

FIGURE 24 - BOLLARD PULL TESTS - REAR WHEELS ONLY, WHEEL PUMPS, 7.50-16 NDCC TIRES, SKIRTS

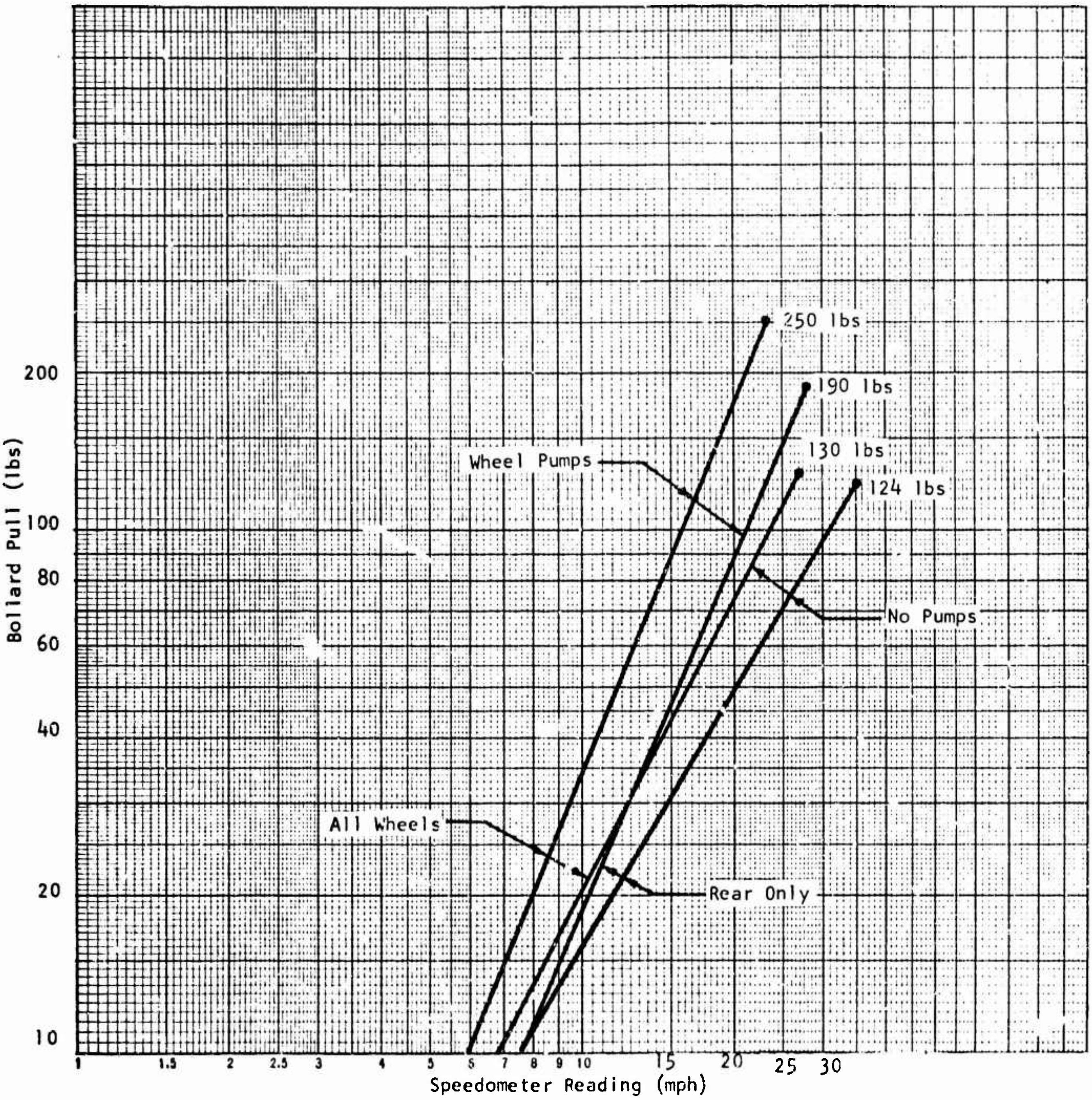

FIGURE 25. CHANGES IN BOLLARD PULL PERFORMANCE USING THE WHEEL PUMPS

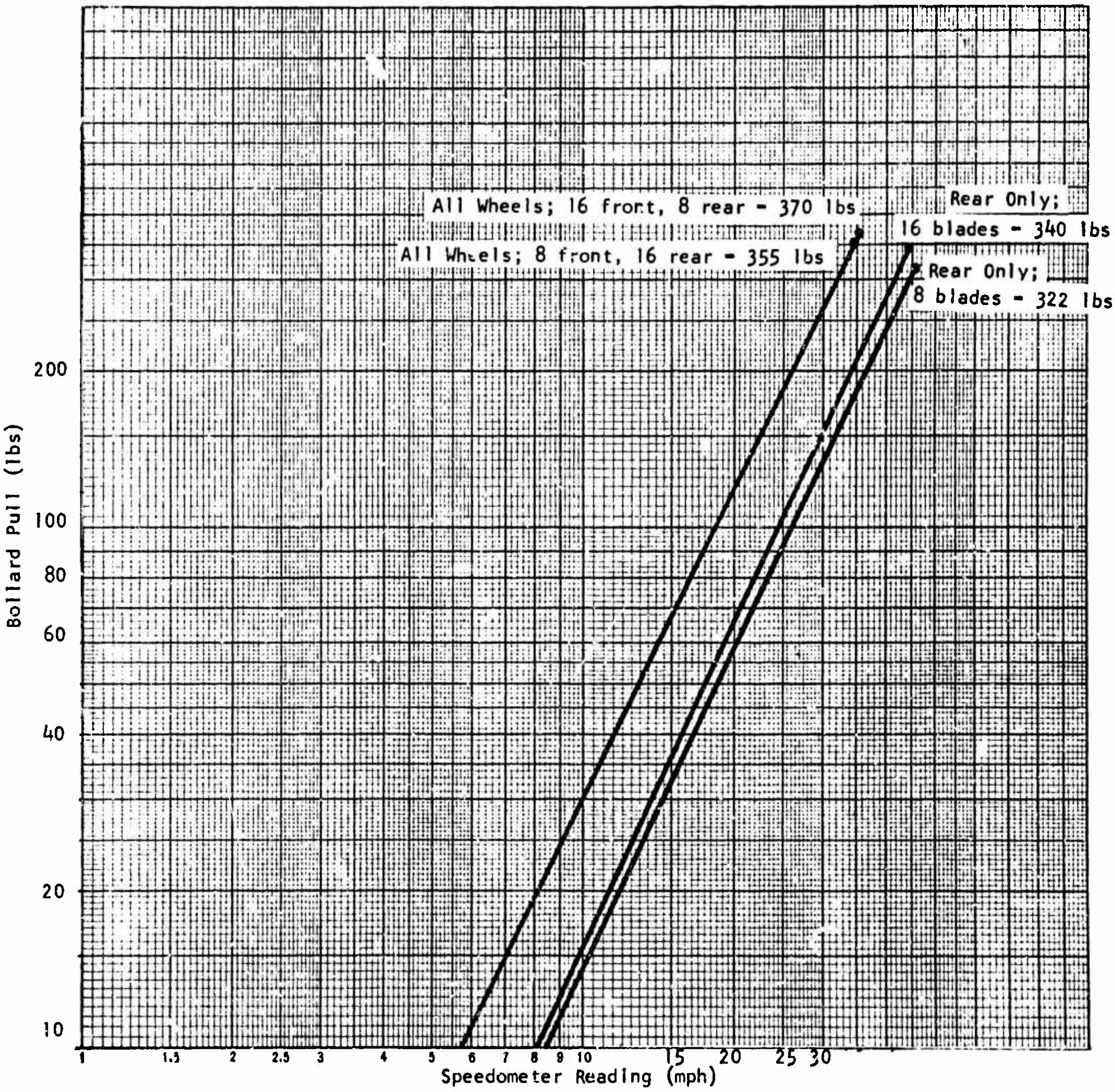

FIGURE 26. SUMMARY OF BOLLARD PULL TESTS - SMOOTH (TREADLESS) 6.50-16 TIRES WITH WHEEL PUMPS FROM FIGS. 27-30

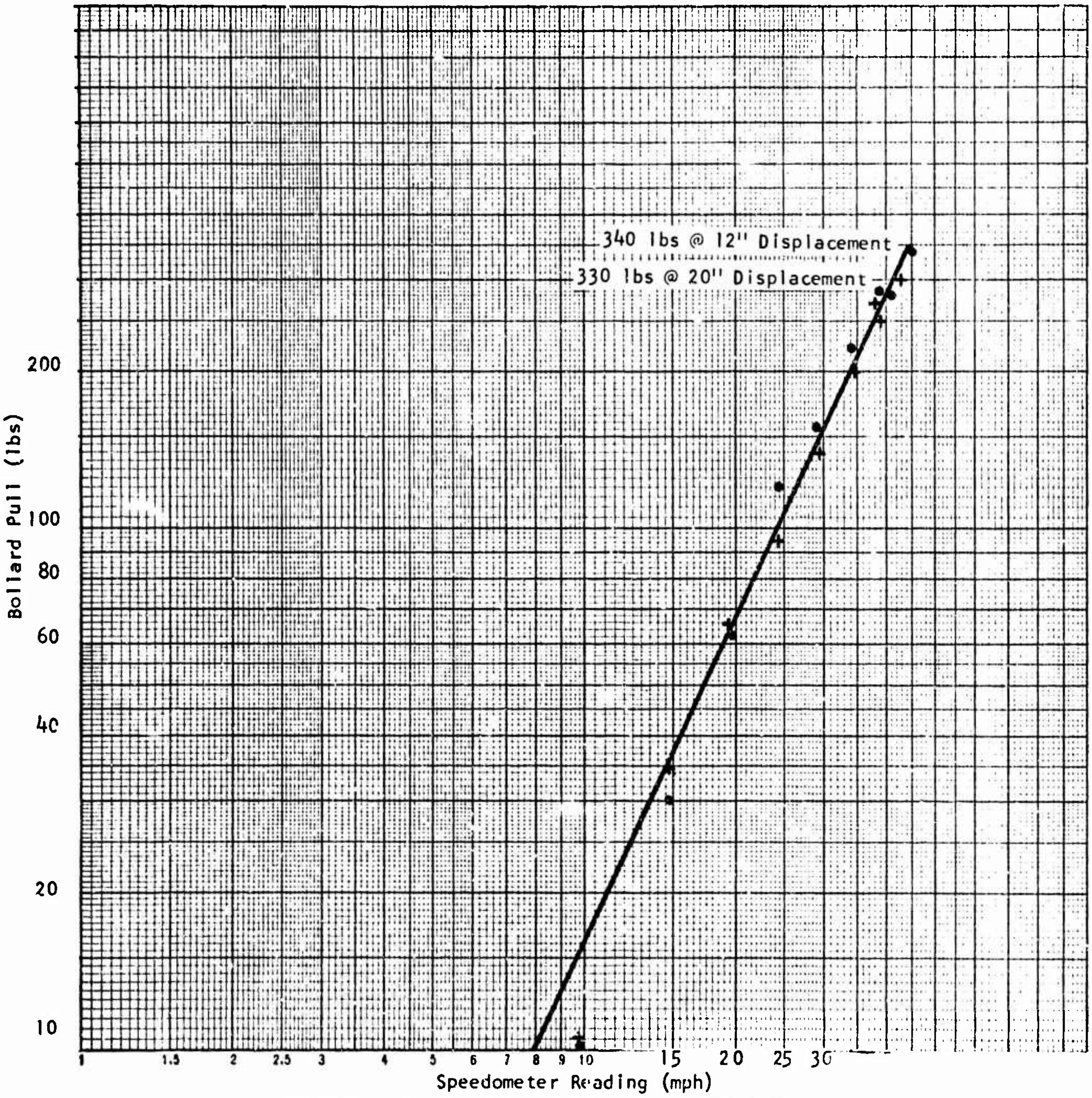

FIGURE 27. BOLLARD PULL TESTS - REAR WHEELS ONLY, 16-BLADED WHEEL PUMPS, 6.50-16 SMOOTH TIRES, NO SKIRTS

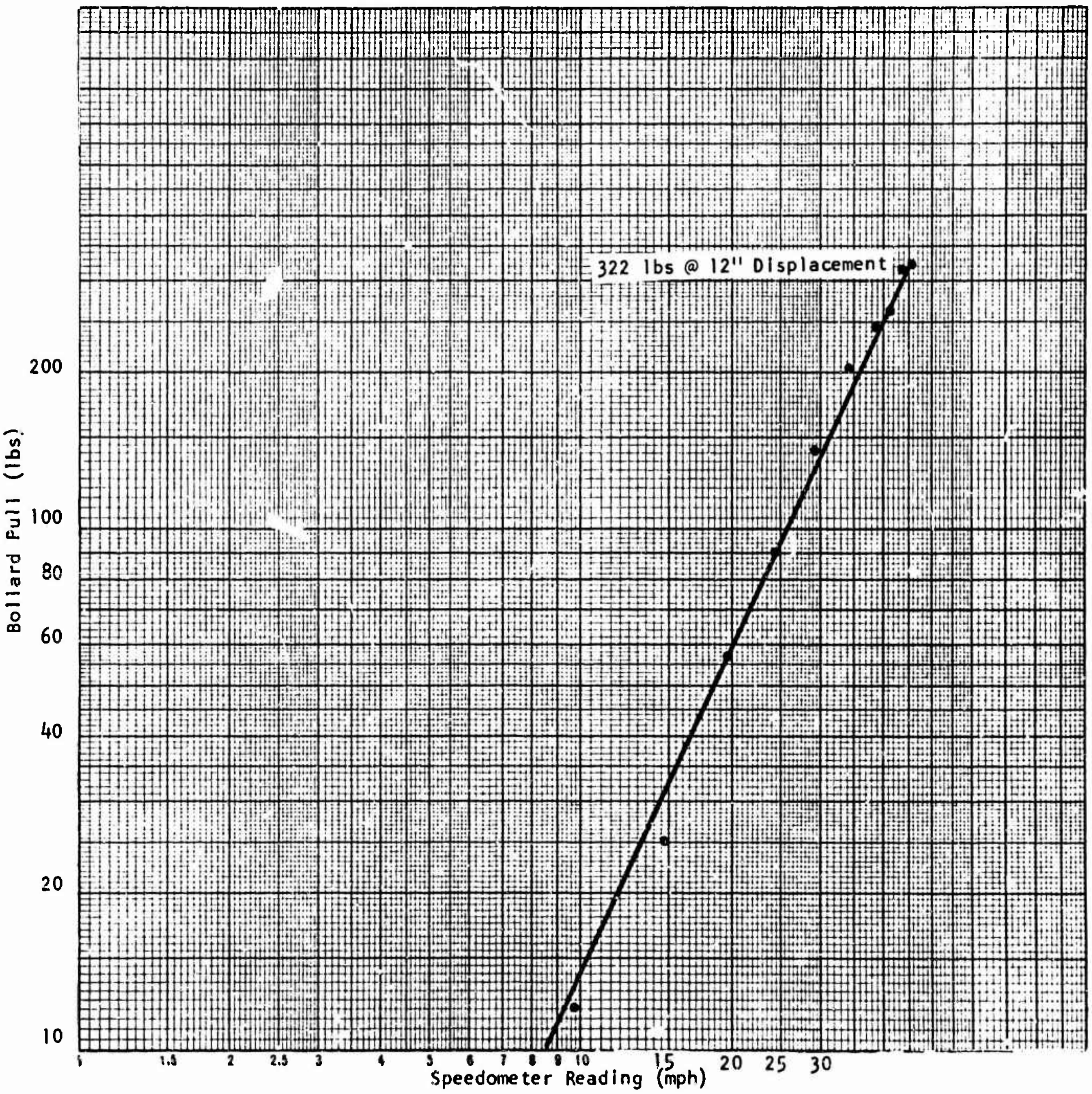

FIGURE 28. BOLLARD PULL TESTS - REAR WHEELS ONLY, 8-BLADED WHEEL PUMPS, 6.50-16 SMOOTH TIRES, NO SKIRTS

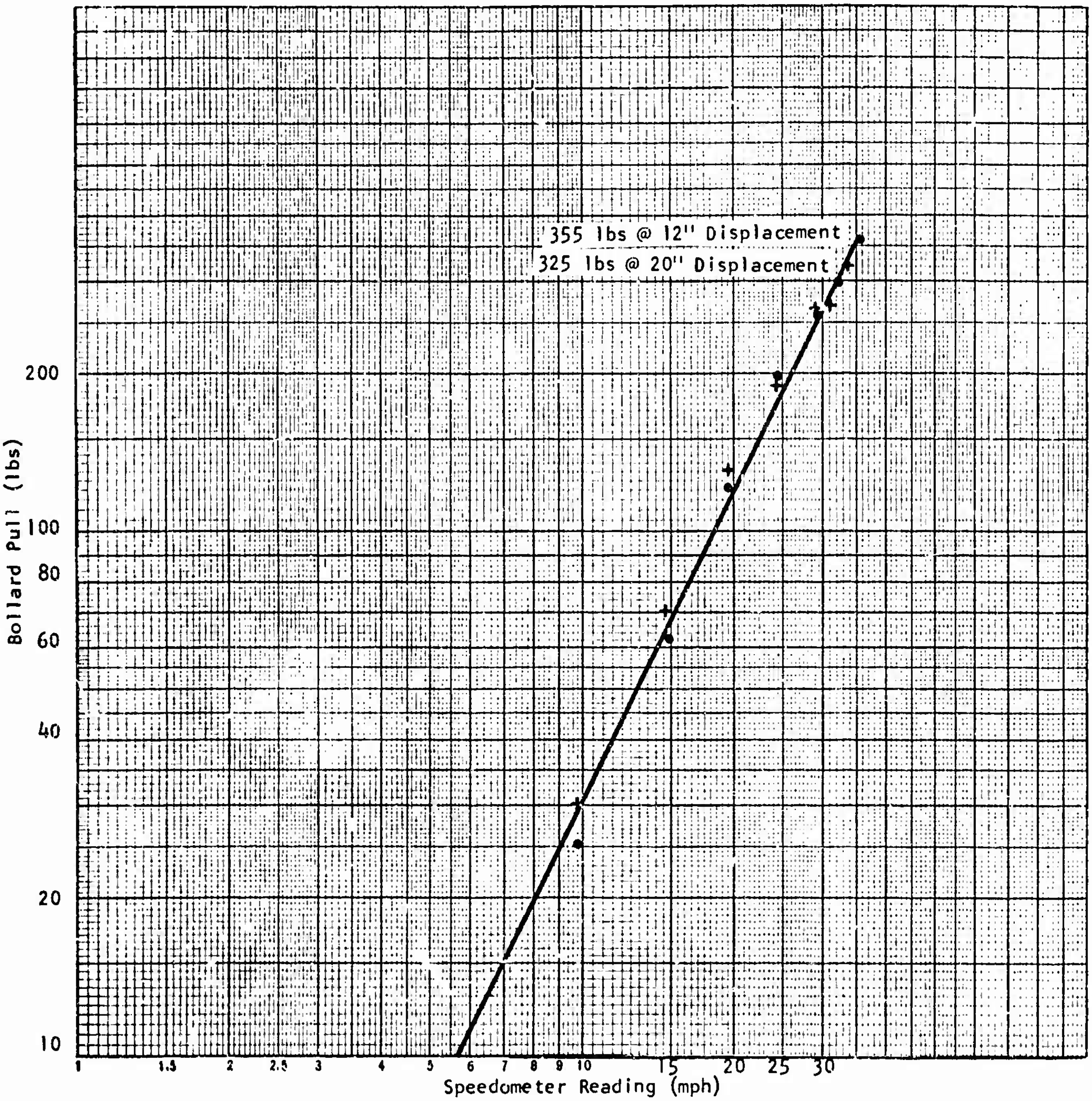

FIGURE 29. BOLLARD PULL TESTS - ALL WHEEL DRIVE, WHEEL PUMPS, NO SKIRTS, 16-BLADED PUMP IN REAR

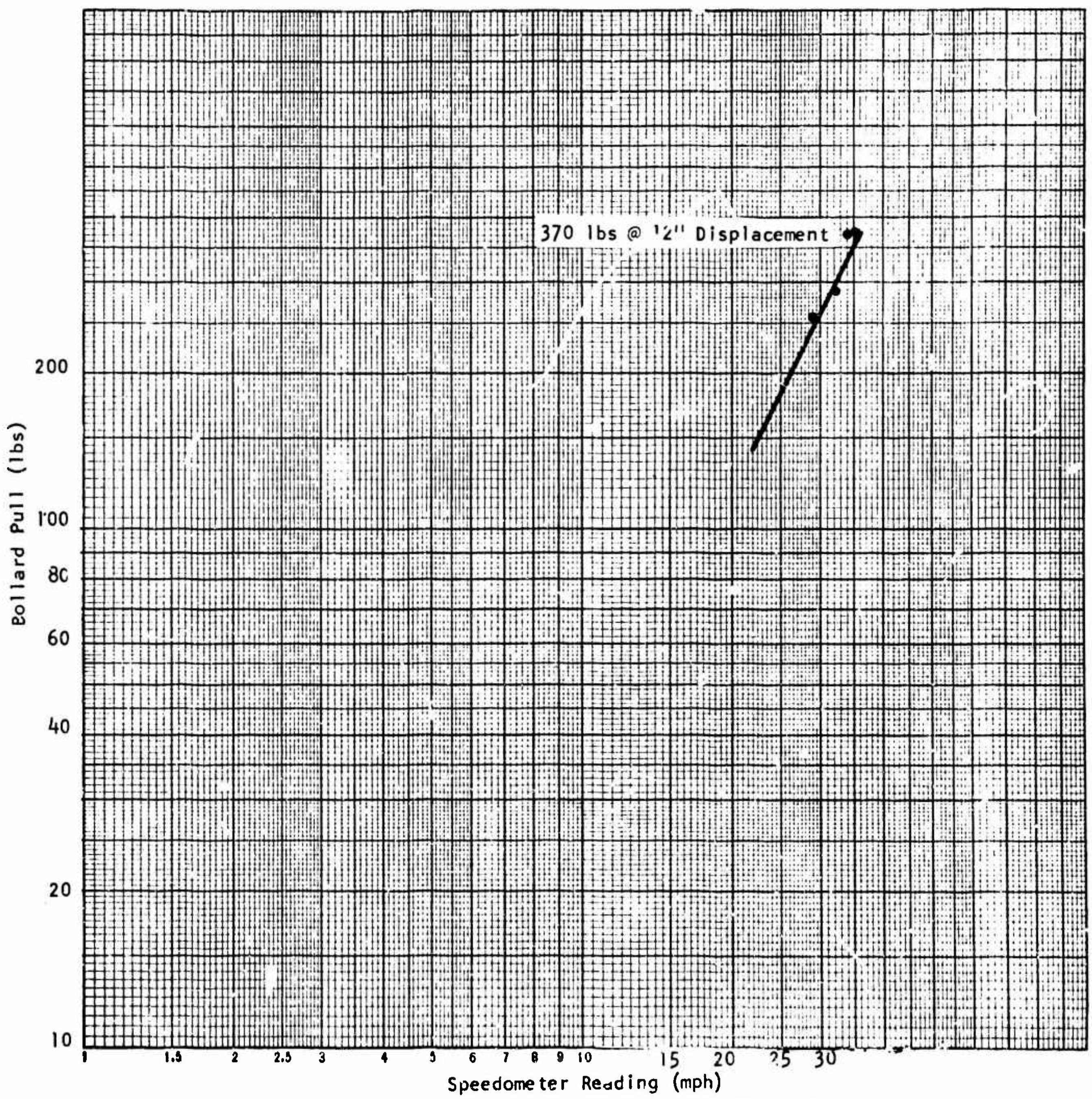

FIGURE 30. BOLLARD PULL TESTS - ALL WHEEL DRIVE,
WHEEL PUMPS, 6.50-16 SMOOTH TIRES
NO SKIRTS, 8-BLADED PUMP IN REAR

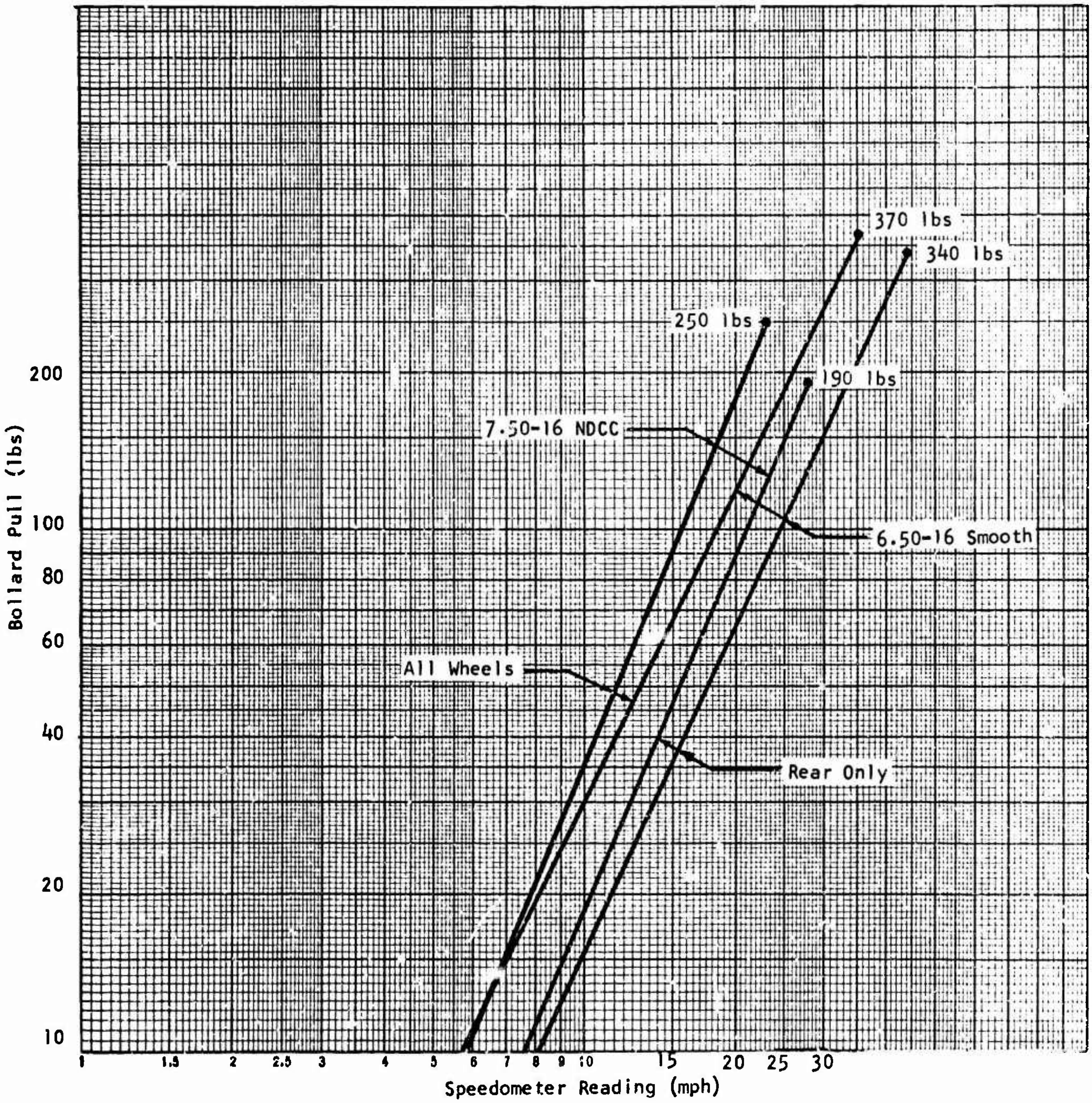

FIGURE 31. BOLLARD PULL TESTS - EFFECTS OF TIRE TREAD ON THRUST

FIGURE 32. COLLECTOR WITH 50% EXIT RESTRICTION NOZZLE

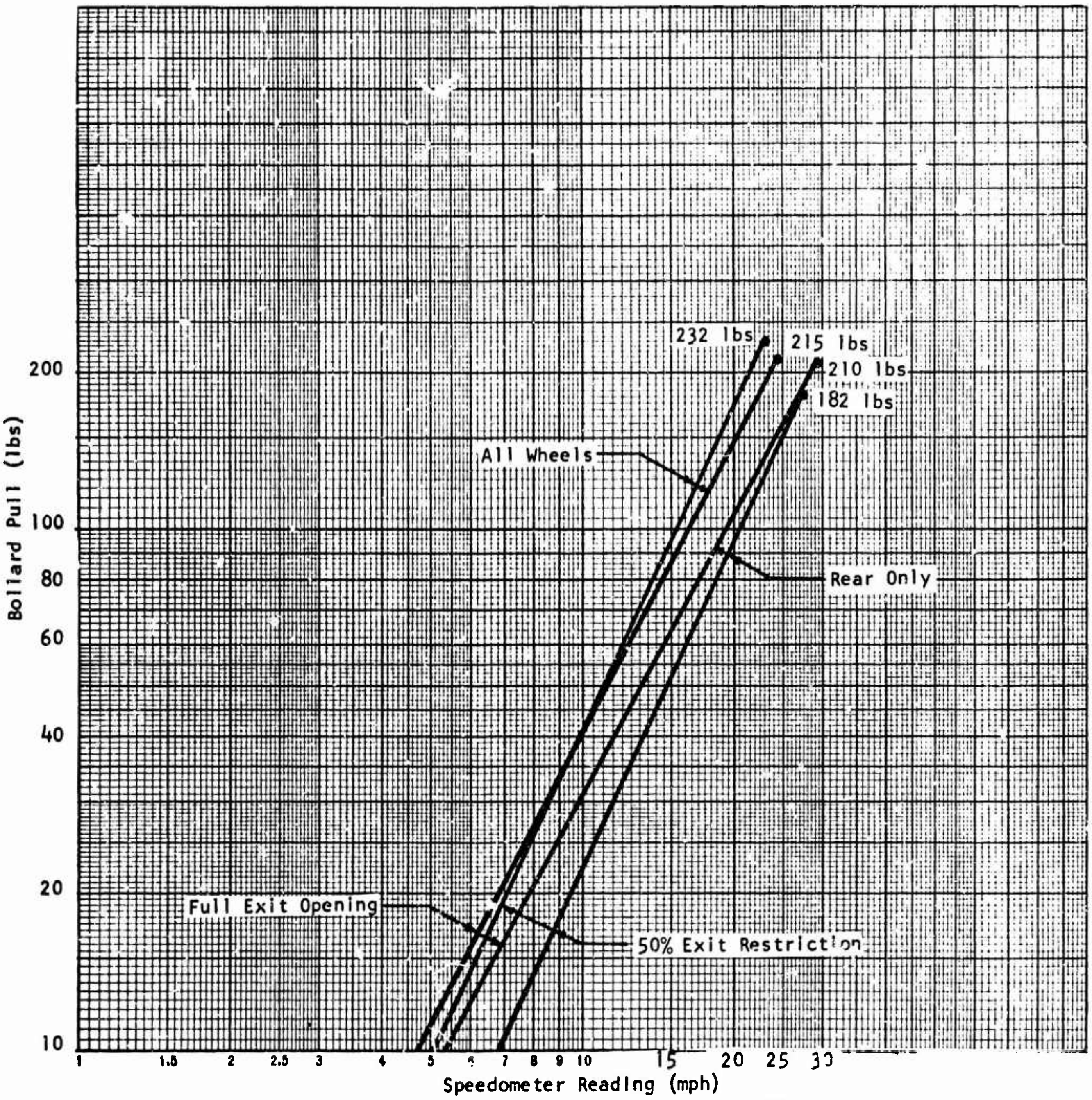

FIGURE 33. SUMMARY OF BOLLARD PULL TESTS - EFFECT OF 50% REDUCTION IN EXIT AREA, FROM FIGURES 20, 34, and 36

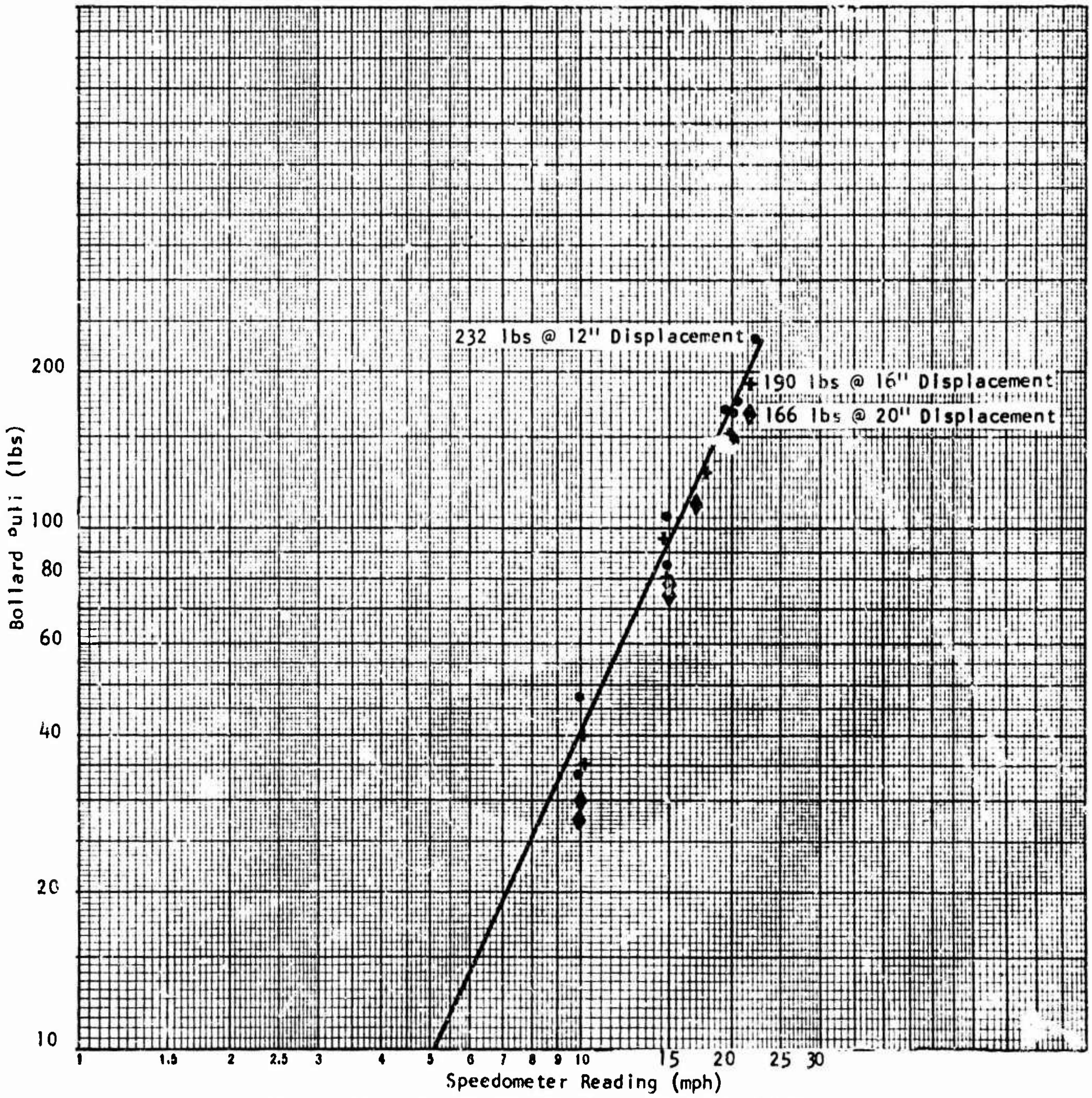

FIGURE 34. BOLLARD PULL TESTS - FOUR WHEEL DRIVE,
WHEEL PUMPS WITH 50% EXIT NOZZLE,
7.50-16 NDCC TIRES, SKIRTS

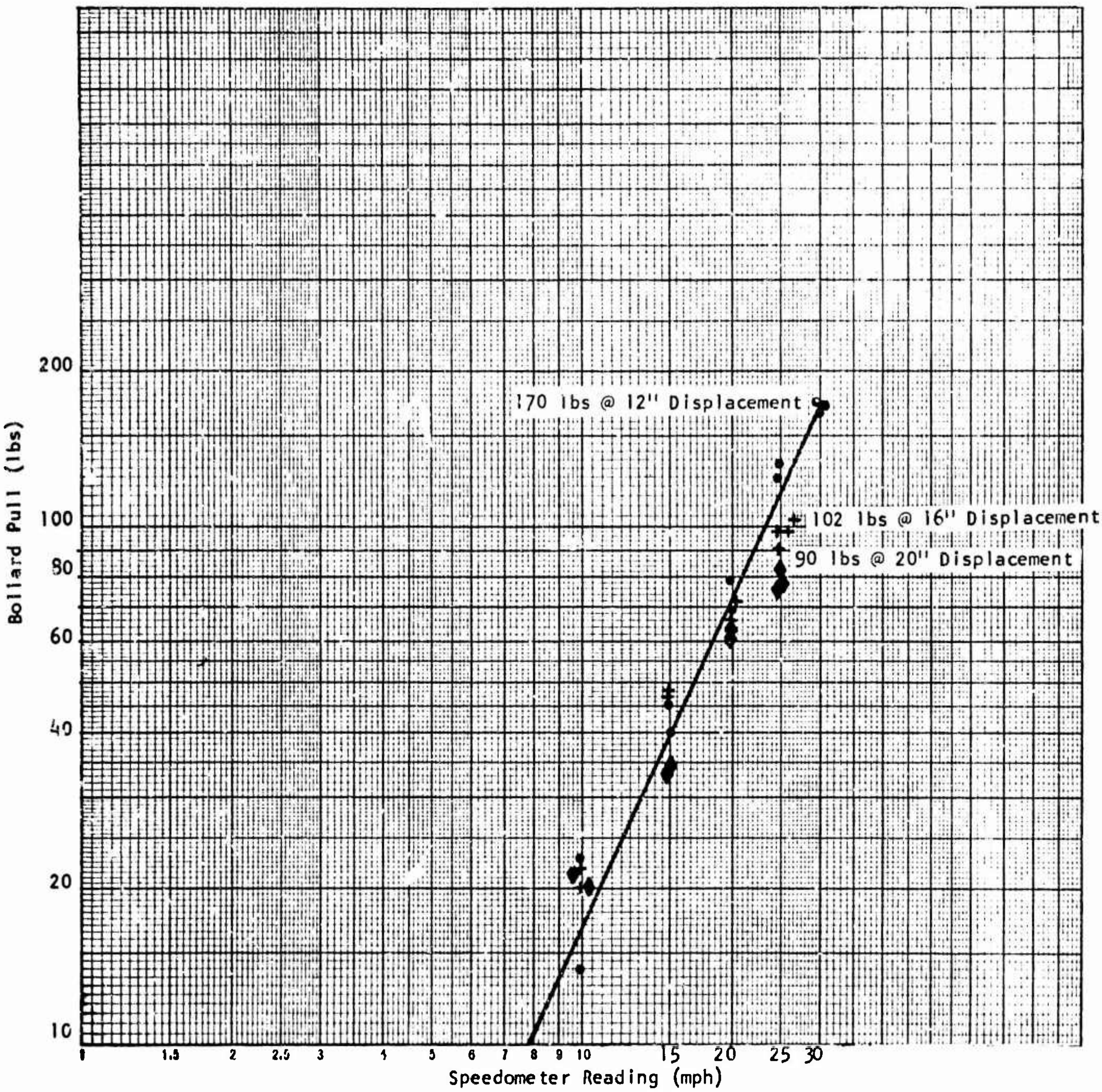

FIGURE 35. BOLLARD PULL TESTS - FRONT WHEEL DRIVE ONLY, WHEEL PUMPS WITH 50% EXIT NOZZLE, 7.50-16 NDCC TIRES, SKIRTS

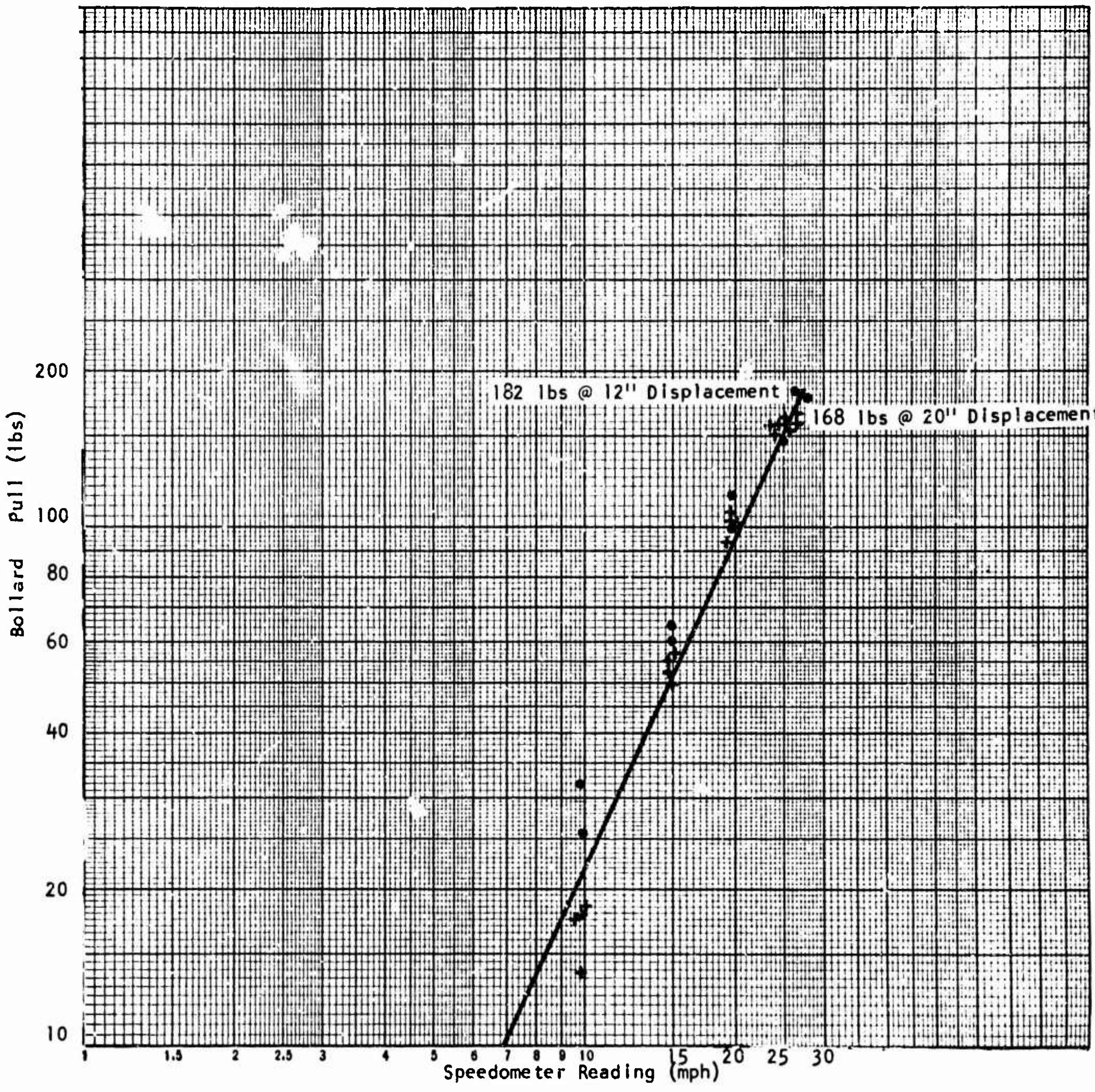

FIGURE 36. BOLLARD PULL TESTS - REAR WHEEL DRIVE ONLY,
WHEEL PUMPS WITH 50% EXIT NOZZLE
7.50-16 NDCC TIRES, SKIRTS

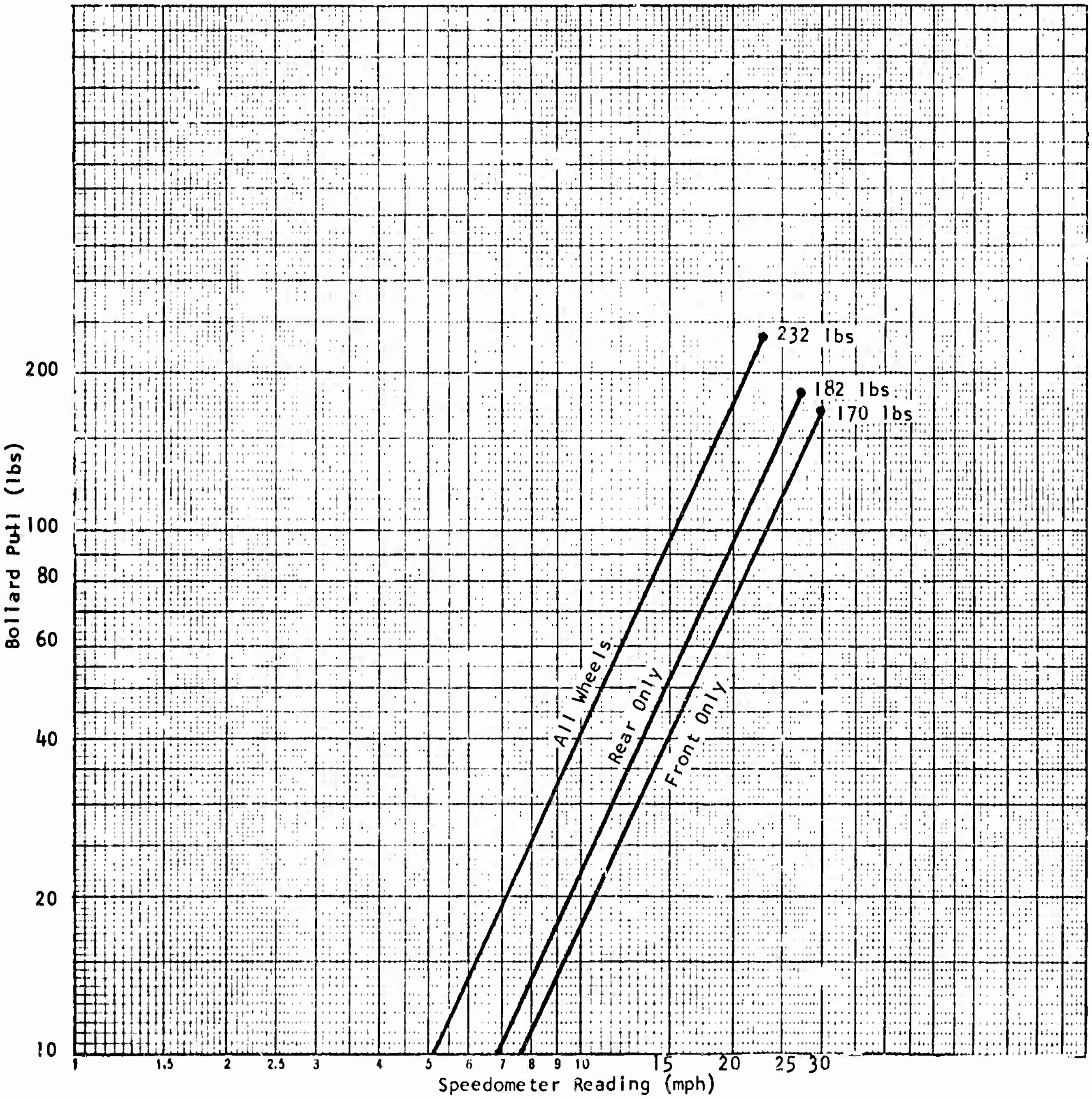

FIGURE 37. SUMMARY OF BOLLARD PULL TESTS - EFFECT OF PUMP LOCATION, FROM FIGURES 34-36

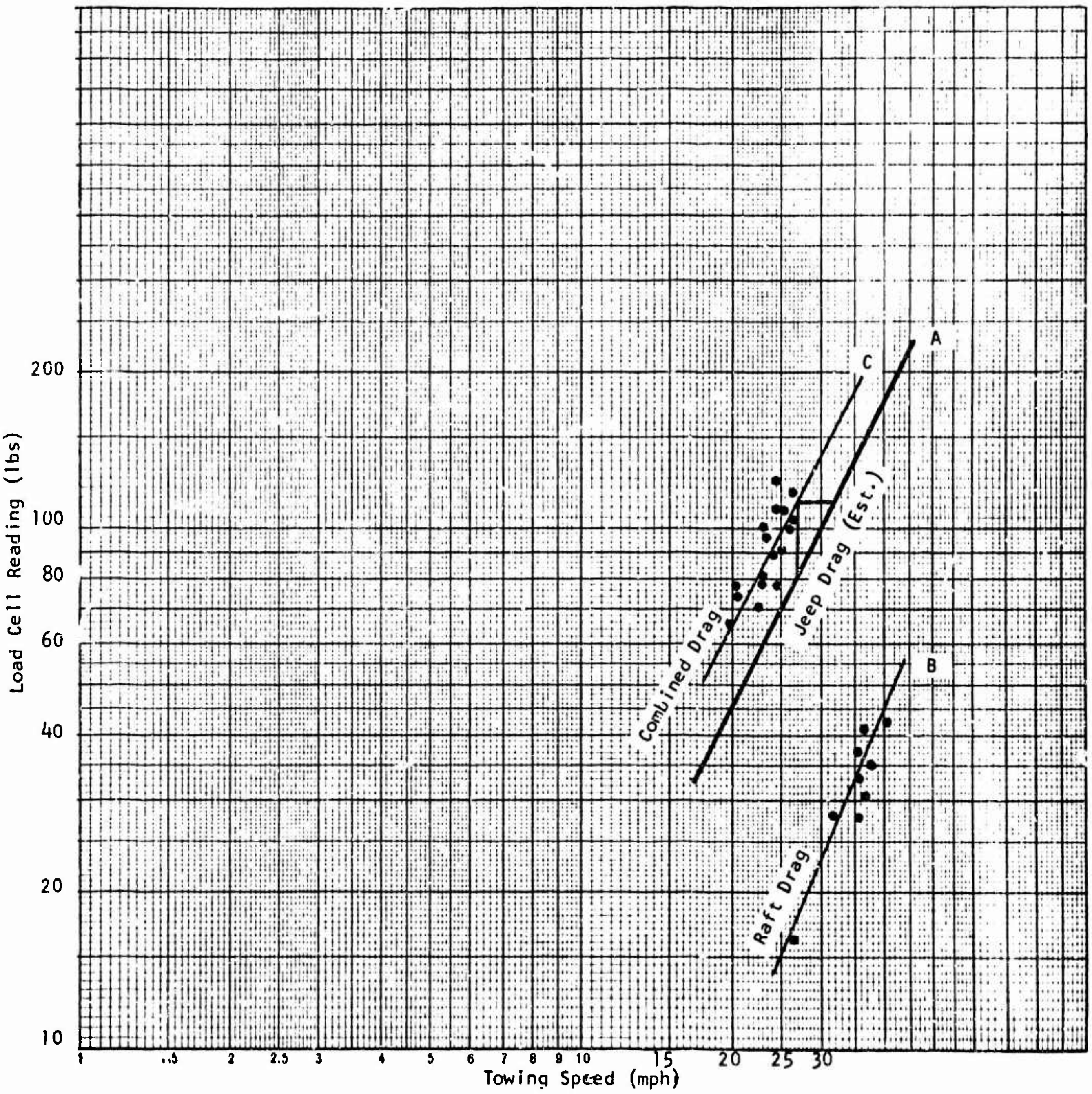

FIGURE 38. FREE RUNNING TESTS

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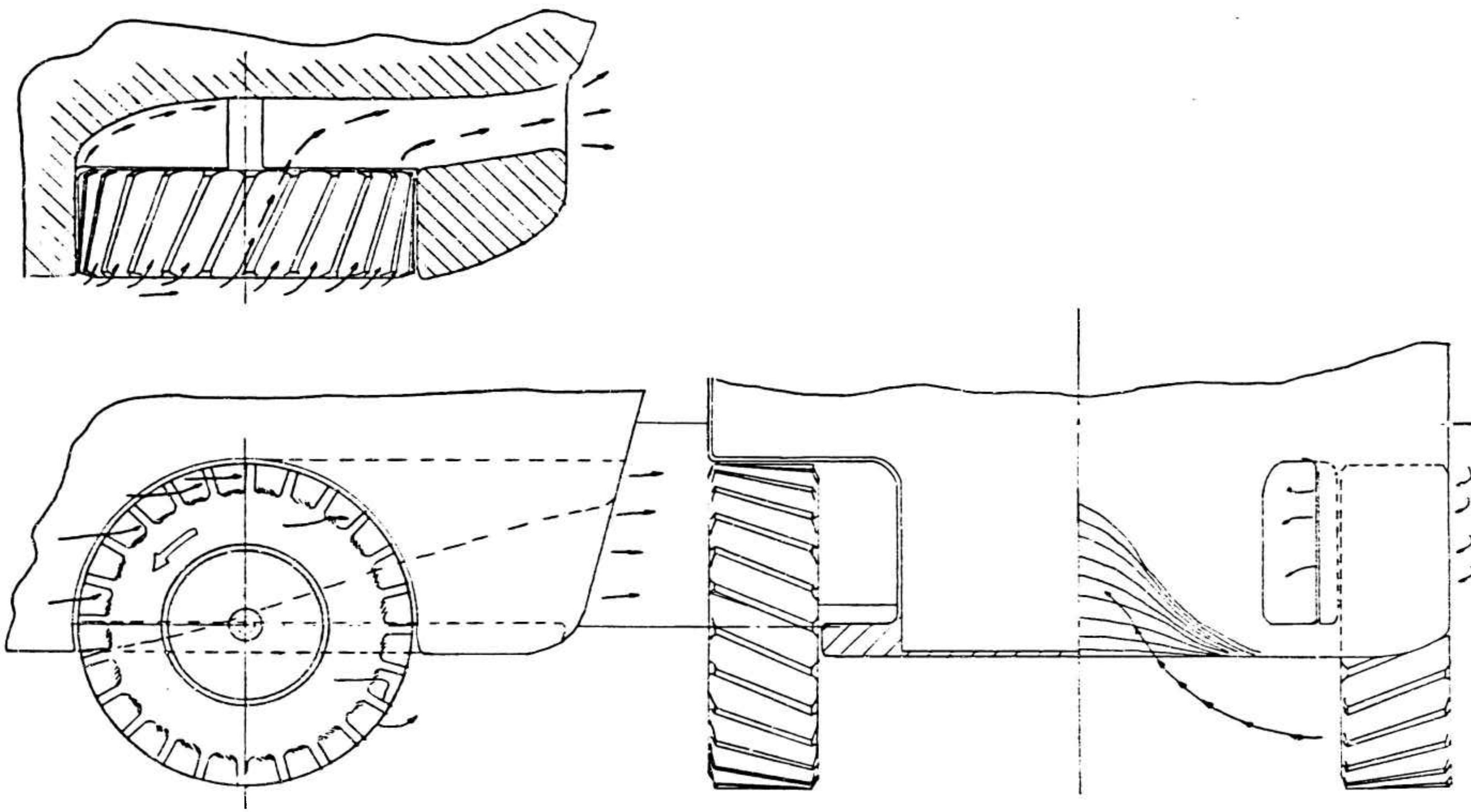

FIGURE 39. CONCEPT SKETCH OF UTILIZING TIRE TREAD PATTERN TO ACHIEVE IMPROVED VEHICLE THRUST

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13. ABSTRACT

A novel propulsion device for an amphibious wheeled vehicle is described. This device, which is an integral part of the vehicle wheels, pumps water between the tire rim and the brake drum inboard into a stationary collector which turns the water rearward, thereby generating forward thrust.

Results of preliminary tests conducted on a stationary pumping system and when mounted on a M151 $\frac{1}{4}$-ton truck are presented.

Tests indicated that the device increases the maximum bollard pull approximately 100% and the maximum speed approximately 40% over propulsion with tires alone. It also materially improves the controllability of the vehicle.

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